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TEXAS A&M TURBOMACHINERY SYMPOSIUM

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A Technical Publication for
Advancing the Practice of
Operating Asset Condition
Monitoring, Diagnostics, and
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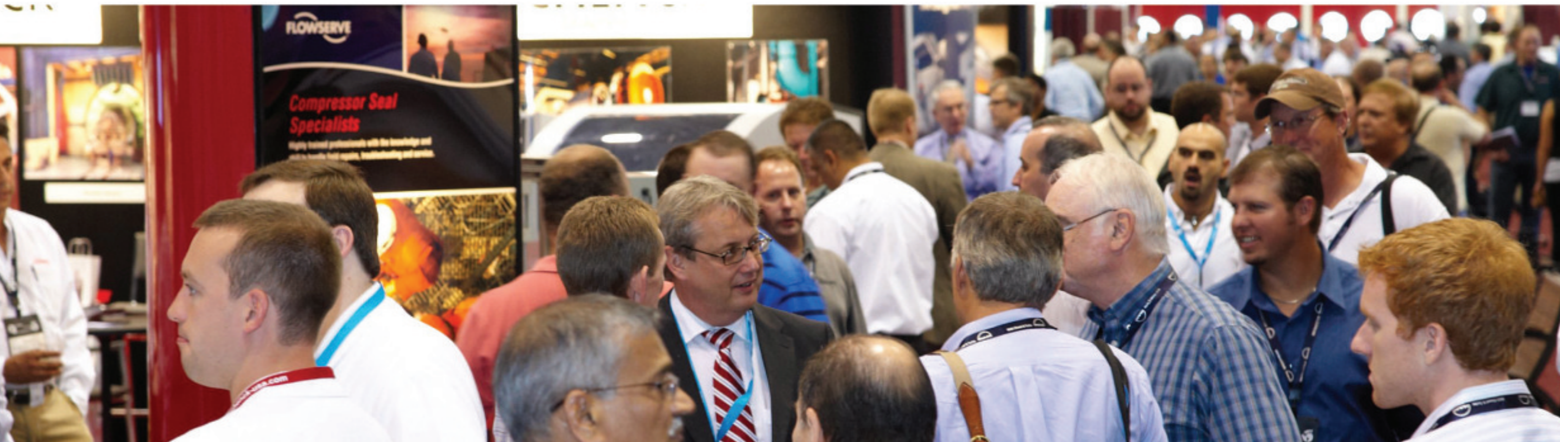

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the Practice of Operating Asset
Condition Monitoring, Diagnostics,
and Performance Optimization

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Editor's Notepad

Gary Swift
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Orbit Magazine
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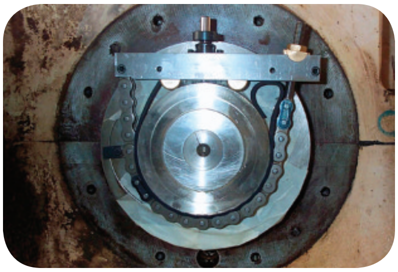
Greetings, and welcome to *Orbit!* While researching the topics for this Turbomachinery-themed issue, I had the enjoyable experience of talking with legacy Bently Nevada employees #3 (Roger Harker) and #4 (Phil Hanifan). Phil retired during the production of this issue, but before he left, he wrote an article for us – looking back on some interesting parts of his 45 years of service.

Roger retired several years earlier, but he has remained in close contact with the Bently Nevada* Asset Condition Monitoring team. Since this issue is focused on Turbomachinery, I asked him about his involvement with the world-renowned symposium hosted by Texas A&M University. It turns out that he participated in 30 consecutive events, starting with the very first one back in 1972. Our conversation quickly moved from "business" to "fun" and Roger shared one of his favorite memories of the event from back in the early days when it was still small enough to be held in College Park (home of the Texas A&M campus).

It turns out that well-known vibration condition monitoring expert, Charlie Jackson, and our own machinery alignment expert, Al Campbell, shared a proud Scottish ancestry. After the day's technical sessions were over, one of their favorite activities was to dress in their traditional kilts, and march through the hotel hospitality suites and around the swimming pool – playing bagpipes and leading an impromptu parade of participants! Newcomers were always taken by surprise, while returning participants who knew what to expect were delighted to join the unofficial festivities once again.

Since this issue focuses on condition monitoring of turbomachininery, we have included articles on several related topics.

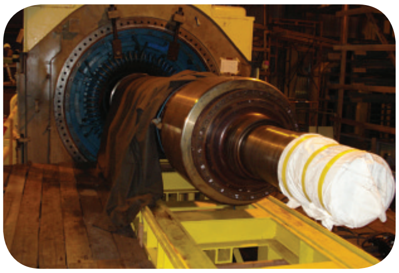


Designing and Machining a Keyphasor* Trigger (page 16)

Our lead article includes some useful tips for successfully field-machining a Keyphasor triggering feature. By using a simple jig to securely support a drill motor, it is possible to machine a shallow circular indentation – or even an elongated slot - in a suitable part of the shaft surface without undue risk. Keyphasor transducers are very important for effective monitoring and diagnostics – and in situations where the OEM has not provided a triggering feature, field-machining of such a feature may be the only option.

System 1* Software Identifies Rotor Instability (page 24)

This article describes a classic instance of fluid-induced instability of a steam turbine driving an air compressor in a fertilizer plant in India. Our diagnostic engineer was able to determine the problem using System 1 data, and plant staff collaborated with the OEM to modify the steam turbine bearings to improve rotor stability. The redesign included machining the bearings from their original cylindrical “sleeve” design to a classic “lemon-bore” design.



Diagnosing Generator Rotor Thermal Sensitivity (page 38)

This case history describes a thermally sensitive generator rotor at a power plant in Thailand. Our diagnostic engineer advised the plant staff as they performed load testing and used other analysis techniques to rule out possible contributing factors. Visual inspection confirmed suspicions of shorted rotor turns, and a partial rewind repair was completed. When combined with a “compromise balance,” these measures allowed the generator to be returned to service until a full overhaul could be accomplished.

2012 has flown by, and it is difficult to believe that this is already the final Orbit issue of the year! We will start the new year with a “back-to-basics” issue on Transducers, and I invite you to join us again in January.

Cheers!
Gary

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GLOBAL TRAINING UPDATE

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- Qatar: Doha
- Russia: Moscow
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NEW ENGINEER TRAINING

The Bently Nevada Technology team is a growing, vital and innovative organization that needs fresh new talent – with new ideas – in order to effectively meet customer expectations. To enhance our team, we hire new Engineers – many of whom have recently completed their university masters or bachelors programs, with degrees in Computer Science or Electrical, Mechanical or Human Factors Engineering.

To facilitate the transition of our new Engineers into the GE organization, we periodically conduct new engineer training classes. These sessions feature presentations from our most senior Engineering mentors, and include topics on our culture, history, Bently Nevada core technology and our product design processes.

Our latest group of new Engineers received training at the Minden, Nevada USA headquarters for the Bently Nevada* Asset Condition Monitoring product line.

FROM LEFT TO RIGHT: Nathan Mayes, David O'Connor (Course Facilitator), Nick Aboumrad, Daryl Van kampen, Joseph Montgomery, Cameron Graybeal, Victor Nunez, Patrick O'Bryan, Jackie Tappan, Daniel Dilts, Phil Hanifan (Mentor), Cindy Hughes (front), Trevor Seyfried (back), Felicia Groso, Gina Morrow and Nick Niemann.

> "IT'S EXCITING TO ADD SUCH GREAT TALENT TO THE GE BENTLY NEVADA ENGINEERING TEAM. BENTLY HAS A PROUD HERITAGE OF TECHNICAL EXCELLENCE AND INNOVATION, AND I KNOW THESE HAND-SELECTED INDIVIDUALS WILL BE ABLE TO CONTINUE THIS LEGACY."
>
> **—ERIC BUTTERFIELD**
> **BENTLY NEVADA**
> **ENGINEERING MANAGER**

> "IT'S ALWAYS A PLEASURE WORKING WITH OUR NEW ENGINEERS, AND HELPING THEM GET OFF TO A GREAT START IN OUR TECHNOLOGY ORGANIZATION."
>
> **— DAVID O'CONNOR**
> **NEW APPLICATIONS**
> **ENGINEER**

Shortly after this class photo was taken, Chief Engineer Phil Hanifan (Bently Nevada employee #0004) retired after 45 years of service. Over this past year, Phil has focused on passing along his wealth of knowledge and providing valuable coaching to our Technology employees before heading to his well-deserved retirement.

At this point it is too early to know for sure, but it is not unreasonable to anticipate that one or more of the new Engineers in this photo may remain with the organization and serve as mentors 45 years in the future – to Engineers from the class of 2057! ■

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THE 3500/45E POSITION MONITOR

Joe Taylor
Bently Nevada Product Manager
josephk.taylor@ge.com

3500E FUTURE EXPANSION
Bently Nevada™
Asset Condition Monitoring
3500/50E TACHOMETER
Rotor Speed
3600
Alert (Over) = 3650 RPM
Danger (Over) = 3900 RPM
3596
Alert (Over) = 3650 RPM
Danger (Over) = 3900 RPM
rature
110
108
108
113
105
103
OK
ALERT
DANGER
BYPASS
MADE IN U.S.A.
MADE IN U.S.A.

3500 ENCORE* SYSTEM, WITH POSITION MONITOR (FIFTH SLOT FROM LEFT). IN THIS EXAMPLE, THE POSITION MONITOR IS CONFIGURED FOR THRUST POSITION MEASUREMENT.

WITH ADDITION OF THE 3500/45E POSITION MONITOR THE 3500 ENCORE FAMILY OFFERS ALL THE MONITOR TYPES FOR A TURBINE SUPERVISORY INSTRUMENTATION (TSI) SYSTEM. THE NEW 3500/45E REPLACES SIX 3300 SERIES MONITOR TYPES (REFERENCE 1):

- 3300/20 THRUST POSITION
- 3300/45 DIFFERENTIAL EXPANSION (DE)
- 3300/46 RAMP DE
- 3300/47 COMPLIMENTARY INPUT DE
- 3300/48 CASE EXPANSION
- 3300/70 VALVE POSITION

Monitor Application

Bently Nevada* position monitors have played an important role in TSI systems since our 5000 Series monitors were introduced back in 1967. TSI systems typically protect steam turbines or gas turbines in the power generation industry, where they are installed to ensure plant safety per regulatory requirements.

In addition to using position-related measurements, TSI systems typically monitor a long list of other machine parameters, including bearing metal temperature, eccentricity, phase, radial vibration, shaft absolute vibration, rotor speed, zero speed & acceleration. Other application-specific electrical and process parameters (generator load, differential current, condenser backpressure, hotwell levels, etc.) are also important inputs to the TSI system for the monitored machine train.

A discussion of these other parameters is outside the scope of this mini-article, but we will briefly summarize the TSI-related functions of the six 3300 monitors listed above.

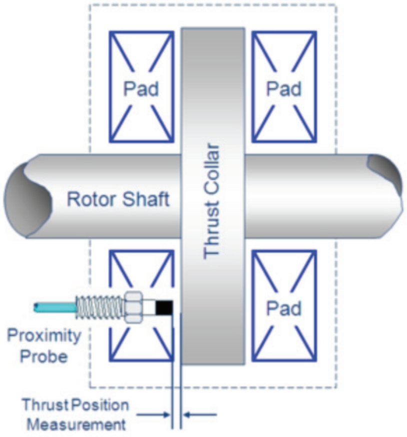
Pad
Pad
Rotor Shaft
Thrust Collar
Pad
Proximity Probe
Thrust Position Measurement

FIGURE 1: When possible, probes are mounted directly in the thrust bearing.

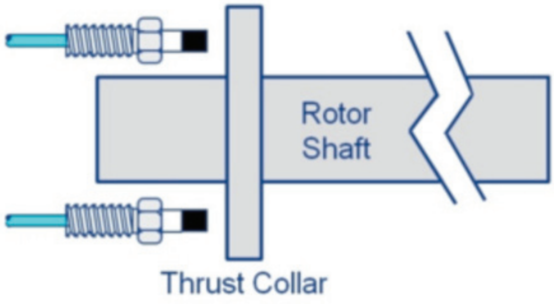
Rotor Shaft
Thrust Collar

FIGURE 2: Typically, two probes are used (Channel A & Channel B).

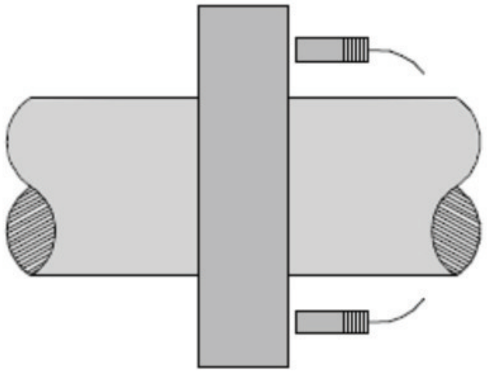

FIGURE 3: Single Input DE measurement.

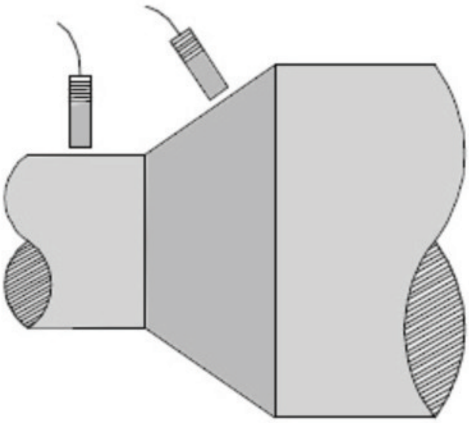

FIGURE 4: Single Ramp DE measurement.

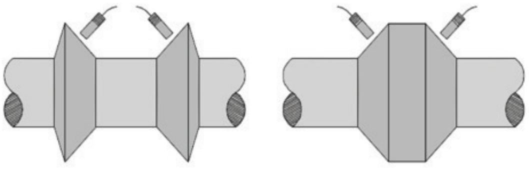

FIGURE 5: Dual Ramp DE measurement

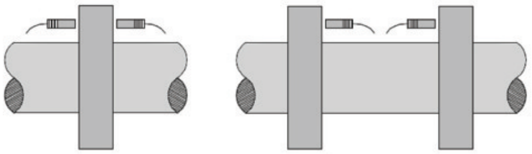

FIGURE 6: Complimentary Input Differential Expansion (CIDE) measurement

Thrust Position

Axial rotor forces are accommodated by thrust bearings that are designed to handle the loading for a normal range of operating conditions. However, if the thrust bearing malfunctions, the rotor can shift axially causing extensive damage when the moving parts contact the stationary parts in the machine. To avoid such problems, proximity transducers and monitoring instruments are used to provide alarms or trip the machine automatically if thrust bearing problems allow the rotor to move to an unsafe axial position.

Thrust Position is a measurement of the position of the thrust collar in the thrust bearing assembly. For this measurement, the relative reference point is the thrust bearing support structure (Figure 1). Typically, two probes are installed in a dual voting arrangement (Figure 2). Such an installation is recommended for maximum reliability.

Differential Expansion (DE)

Large steam turbines (and some gas turbines) exhibit non-uniform thermal growth rates of the rotor and casing during machine startups. If it is not carefully monitored and controlled, excessive differential expansion can cause the spinning rotor and stationary casing to rub, with catastrophic results. This measurement uses proximity transducers to measure the difference between the relative positions of the machine rotor and the machine casing as they grow in length from cold conditions to normal operating temperatures.

- Single Input Differential Expansion – Uses a single probe to measure DE in the axial direction. It is very similar to Thrust Position measurement, except the probe is mounted to the moveable turbine casing, rather than to the fixed thrust bearing. Another important difference is that the DE probes are purposely installed at a significant distance away from the thrust bearing, in order to observe the maximum possible thermal growth of the rotor away from the thrust bearing. The collar in the simplified drawing of Figure 3 is not the thrust collar, but rather, a suitable axial target at some distance away from the thrust bearing. As shown, two probes can be used to provide redundancy in case one probe fails.
- Single Ramp Differential Expansion – Increases effective probe linear range by observing a sloped rotor surface, instead of a perpendicular collar. This application requires a second probe to measure radial vibration for compensation of the DE signal (Figure 4).
- Dual Ramp Differential Expansion – Similar to the single ramp measurement, only BOTH probes observes the sloped rotor surface. Again, compensation is performed to remove the effects of radial vibration from the DE signal (Figure 5).
- Complimentary Input Differential Expansion – An extension of the Single Input measurement, this application combines the signals from two separate axial probes to effectively double the linear range over that of a single transducer (Figure 6). As the target nears the end of the usable range for the first transducer, it enters the range of the second transducer.

Case Expansion

Case expansion is the thermal growth of the machine casing as it expands during machine startup and loading operations. The case expansion transducer system provides information on the growth of the machine case relative to the foundation. The sensors – most often Linear Variable Differential Transformers (LVDTs) – are typically mounted on the foundation at the opposite end from where the turbine casing is attached to the foundation (Figure 7). This allows measuring the full extent of the casing thermal growth.

If sliding feet become jammed and prevent the casing from expanding freely, it can warp, which misaligns the bearings and shaft seals. Such conditions greatly increase the possibility of a rub between the rotor and the stationary components that are mounted to the inside of the casing.

Note: LVDT output varies with position of the movable core that links the magnetic flux from the primary to secondary coils.

Valve Position

A complete TSI system for steam turbine generators normally includes a measurement of the position of governor valves to assist in determining steam flow. Valve position thus provides an indication of turbine load while the generator is online. Correlating the valve position with other process measurements, such as steam temperature, is useful in determining overall turbine operating efficiency. We have provided valve position monitoring capabilities ever since our 7000 Series monitors were introduced in 1972.

These monitors are typically set to display valve position as percent open or percent closed. The measurement is based on the full linear range of motion for an individual valve or group of valves, or the full angular range of a rotating camshaft that operates numerous valves in an established sequence. Historically, the transducers used for making valve position measurements have been either LVDTs or rotary potentiometers.

LVDT Position Measurement

For steam turbine throttle (governor) valves that use linear actuators, an LVDT is ideal for measuring the valve position. AC LVDTs do not require temperature-sensitive demodulation circuitry to convert the AC output signal to a DC value, so this is the type of LVDT that is most commonly applied in high temperature areas that exist near the hot steam supply lines and valves ("DC" LVDTs are more typically used for Case Expansion measurements).

For this application, the LVDT is mounted to the foundation or fixed part of the hydraulic actuator, while the plunger is connected to the moving part of the actuator (Figure 8). A hydraulic valve actuator can either operate a single valve or several valves in sequence by lifting the end of a beam to which several valve stems are attached. For such a ganged design, valve position can be monitored by measuring position of the beam.

Rotary Potentiometer Position Measurement

Some actuator designs use a rotating camshaft to operate the governor valves. For this type of system, a rotary potentiometer may be more appropriate than an LVDT, as it can measure the angular position of the camshaft directly (Figure 9). Inside the potentiometer, the rotating shaft is attached to a conductive brush or "wiper" that contacts a semi-circular resistance path – acting as a voltage divider. Output voltage is proportional to the angular position of the valve actuation camshaft, which correlates to the position of the valves. ■

References

1. Bently Nevada Specifications and Ordering Information, 3500/45E Position Monitor, Part Number 289792-01

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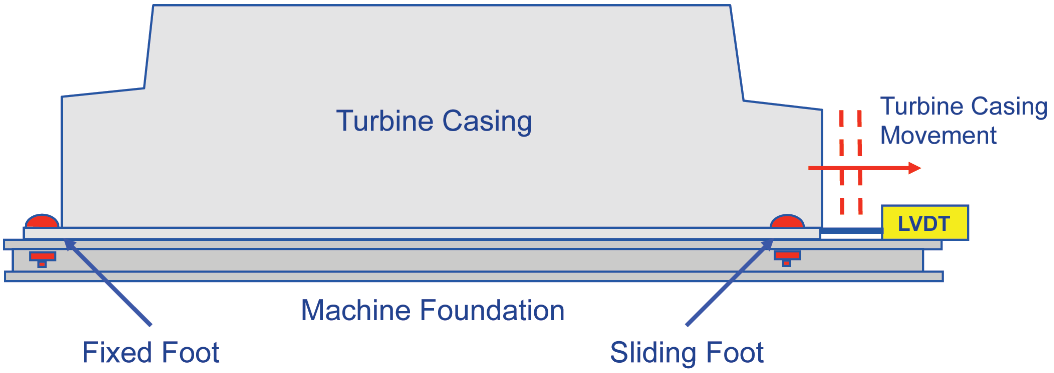

Turbine Casing
Turbine Casing Movement
LVDT
Machine Foundation
Fixed Foot
Sliding Foot

FIGURE 7: LVDT(s) installed to measure expansion at the sliding end of the machine casing. By using two LVDTs, it is possible to detect a situation where one sliding foot has become jammed, but the other one is moving normally.

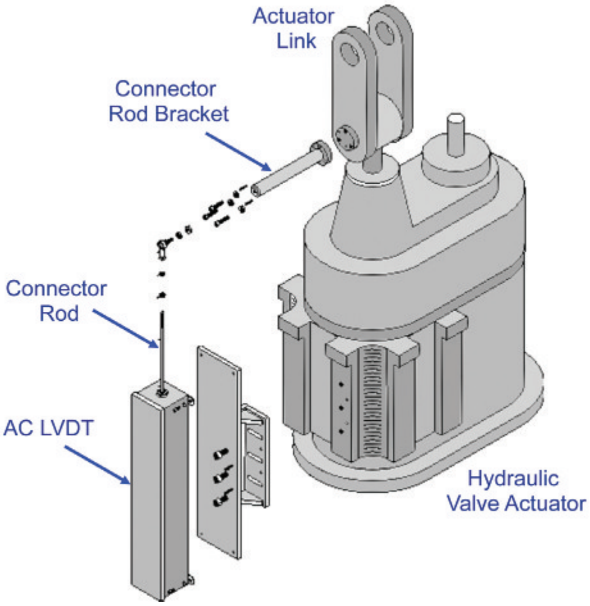

Actuator Link
Connector Rod Bracket
Connector Rod
AC LVDT
Hydraulic Valve Actuator

FIGURE 8: This simplified drawing shows how an LVDT can be used to measure valve positioning opening based on the linear position of the hydraulic actuator.

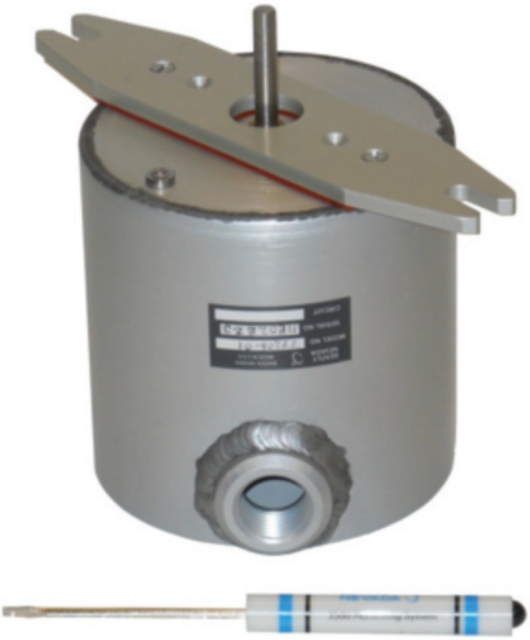
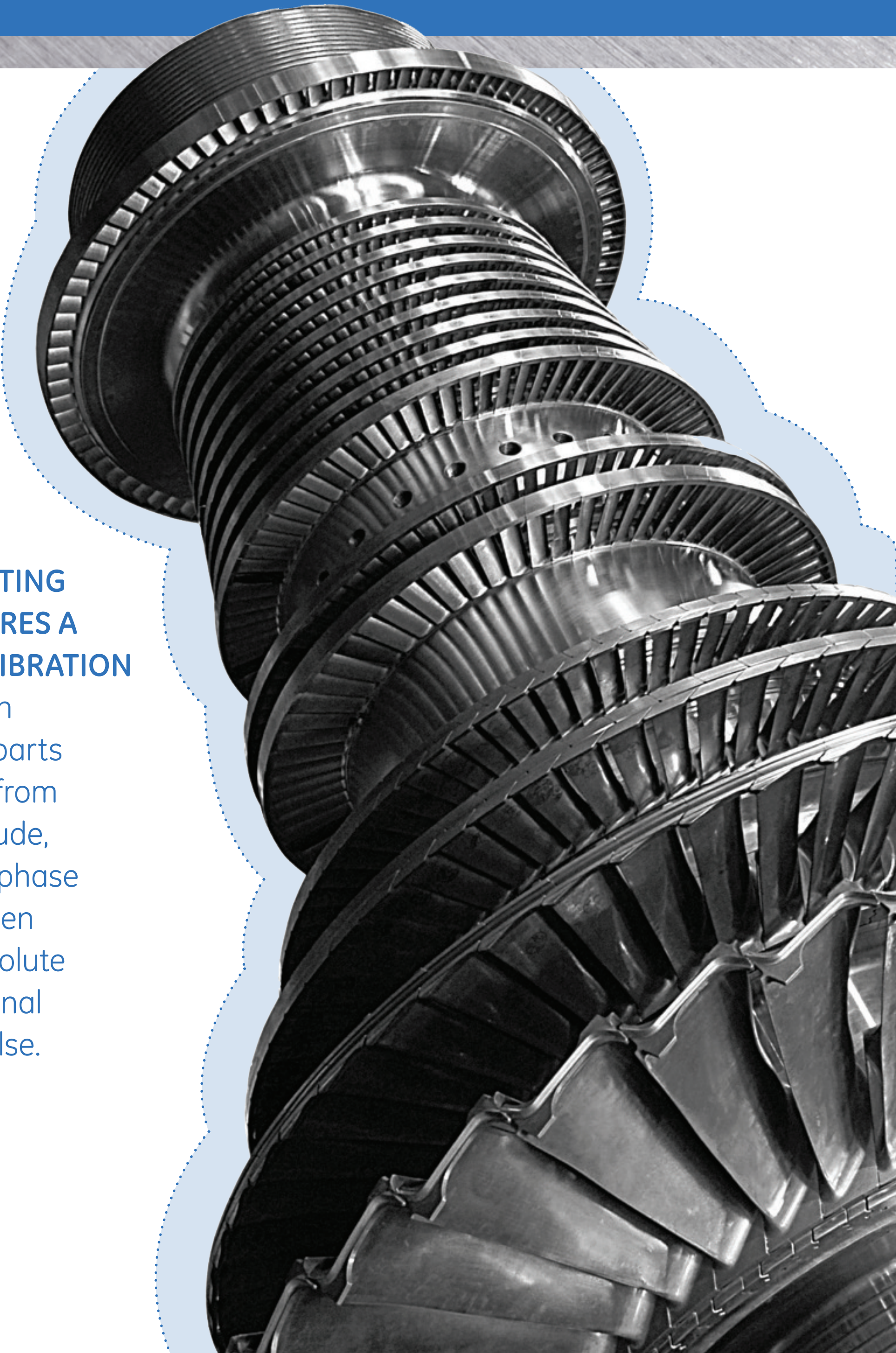


FIGURE 9: Rotary Potentiometer, with a small 135 mm (5.3 in) long screwdriver for size reference. The shaft that is visible extending from the top of the potentiometer housing is attached at the end of the camshaft, and rotates as the camshaft rotates.

DESIGNING AND MACHINING A KEYPHASOR* TRIGGER

Effective Site Machining Options



ACCURATE DIAGNOSIS OF ROTATING MACHINERY PROBLEMS REQUIRES A COMPLETE SET OF MACHINE VIBRATION INFORMATION. This information comes from the three primary parts of the data that we can obtain from a vibration signal: direct amplitude, frequency, and phase. Relative phase is the timing relationship between two vibration signals, while absolute phase compares a vibration signal to a once per-turn reference pulse.

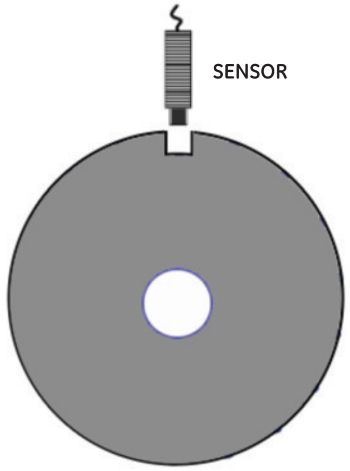


FIGURE 1: This example shows an appropriate triggering feature that provides a once-per-turn reference pulse.

The Keyphasor signal is a once-per-turn voltage pulse provided by a transducer (normally an eddy current proximity measurement system). The Keyphasor signal is used by monitoring, diagnostic, and management systems to generate filtered vibration amplitude, phase lag, speed, and a variety of other useful information. Keyphasor-referenced information can help the Operator or Machinery Specialist identify developing machine problems or distinguish serious problems from less serious ones.

The Keyphasor signal is used to generate more than one third of the vibration-related information regarding the condition of the machine. Phase (relative and absolute) is a critical part of this information. Without phase information, overall machine condition and machine faults would often be very difficult, if not impossible, to diagnose (Reference 1).

What is a Keyphasor Trigger?

The Keyphasor measurement requires a physical feature, such as a coupling keyway or an elongated notch that can provide a once-per-turn event trigger for the signal pulse (Figure 1). The system includes a proximity probe, extension cable and Proximitor* Sensor. This application is required in order to provide Bently Nevada* and other machinery protection and monitoring systems with a once-per shaft rotation phase reference pulse.

Although some machines already have an appropriate triggering feature provided by the manufacturer, others do not. Proper machining of the required shallow triggering feature (usually a circular hole or elongated notch) is sometimes overlooked in Keyphasor retrofit installations.

For effective asset condition monitoring and protection, the monitor systems need to have a reliable once-per-turn triggering event for each major machine component (driver, gearbox, compressor, etc.) in the monitored machine train. Convention recommends that the prime Keyphasor event is located on the driving unit.

Some machine trains (such as a power-recovery train) can have two possible drivers, depending on the operating mode of the plant. Such designs may have a separate Keyphasor installation on each of the drivers. For convenience in establishing an accurate reference, the angular location of the triggering feature is often aligned to a significant rotor feature, such as #1 coupling bolt, designated balance hole or plane or a drive keyway.

As shown in Figure 2, the recommended practice for machinery protection and condition monitoring systems is to locate redundant Keyphasor transducers viewing the same once-per turn triggering feature. This redundancy ensures that a reliable Keyphasor signal will be available even if one transducer system fails. Figure 3 shows an actual probe installation.

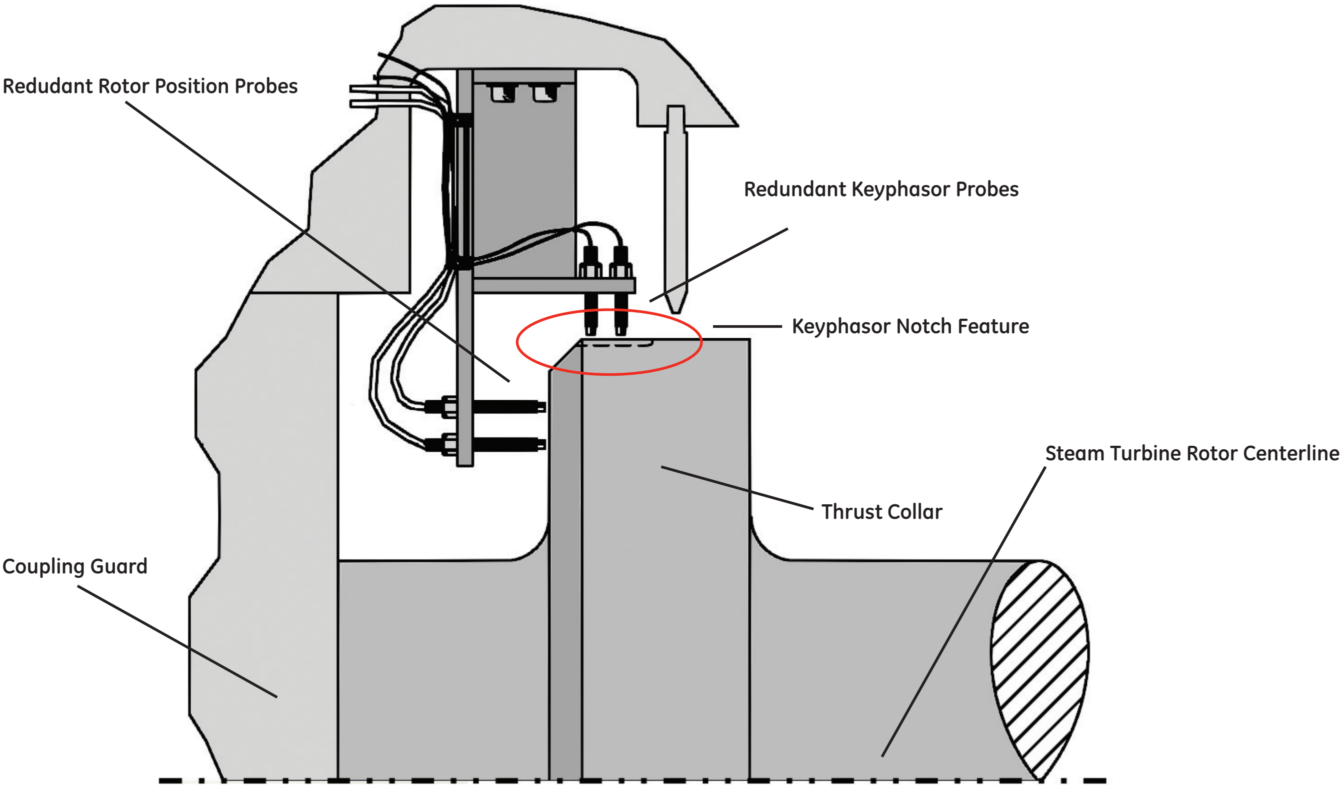

Redudant Rotor Position Probes
Redundant Keyphasor Probes
Keyphasor Notch Feature
Steam Turbine Rotor Centerline
Thrust Collar
Coupling Guard

FIGURE 2: This simplified installation drawing shows redundant axial rotor position probes observing the flat surface of the thrust collar, and a pair of Keyphasor probes observing a notch feature in the outer circumference of the collar. Note: It is important that specified clearance is maintained between adjacent probes in order to avoid interference.

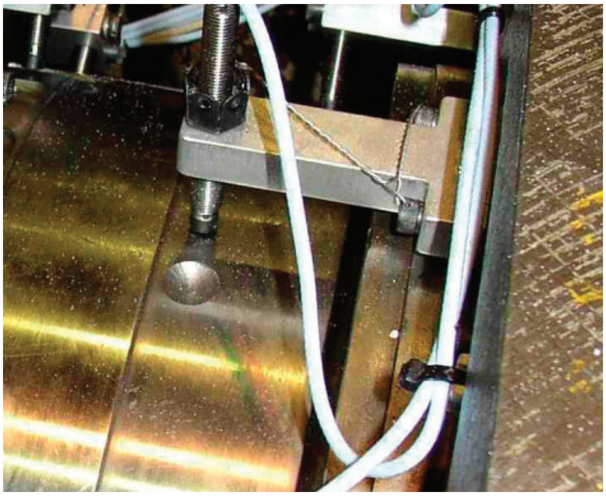


FIGURE 3: Example installation: Triggering feature (circular indentation) is visible in the shaft surface just below the Keyphasor probe. Observe the recommended practice of using safety (lock) wire on the probe nut and bracket mounting bolts, as well as cable anchors securing the probe cable.

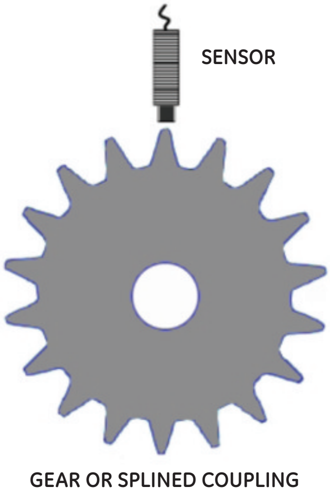

SENSOR
GEAR OR SPLINED COUPLING

FIGURE 4: Multiple features provide multiple events per rotation, and are NOT appropriate for triggering a phase reference signal.

A Question of Balance

When planning to machine a triggering feature for a Keyphasor retrofit application, engineers sometimes become concerned about deliberately creating a potential source of out-of-balance and often want to consider a counterbalance located at 180° opposite the designed feature. This second feature, if located in the same lateral location on the shaft gives rise to the system seeing two events per turn, hence creating application issues. However, it is generally feasible to locate a section of rotor not forming part of a journal, seal area, probe track or lying in the Keyphasor lateral plane, that can accommodate the location of such a counterbalance.

Any form of machining that is performed on a machine shaft has to be carried out with due regard to the integrity of the unit. The volume of material removed to form a Keyphasor triggering feature should be produced under controlled conditions that can be replicated to give an identical effect to create a counterbalance.

While it is conventional to refer to out-of-balance in grams of mass, it is sometimes easier to calculate the material in volume, at a given radius from the shaft centerline to the center of the removed volume. Should the required counter balance hole need to be located on a larger or smaller diameter part of the rotor, the volume & radius method provides an easier calculation. This is true even when there are differing shaft materials involved, although caution should be exercised, as there are some instances where material differences can have an effect.

A question that should be asked is whether a counterbalance is actually needed. The possible need for a counterbalance may be more applicable to smaller, higher speed machinery. On large units such as steam or gas turbine generator machine trains, it may well be found that there is no need to provide such a counterbalance – as the unbalance effect of the removed material is miniscule compared with the huge overall mass of the machine rotor.

Multi-Event Measurements

We have sometimes seen examples where engineering companies and OEMs provide their machines with designs to install a Keyphasor probe to observe the teeth on the gearwheel in a gearbox or on a multi toothed speed sensing wheel – assuming that an electronic multiplier/divider could be utilized to provide a once-per-turn signal (Figure 4). However, such an application does not fulfill all of the requirements of a repeatable once per turn phase reference event and should therefore be avoided. A multi-toothed design is appropriate for Speed, Zero-Speed and Overspeed applications, but it should not be used for phase reference.

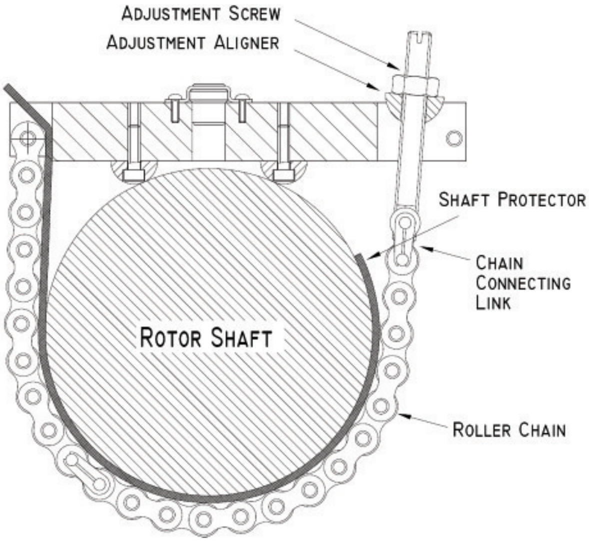


FIGURE 5: Simplified end-view drawing of a useful drilling jig. The machining fixture is the horizontal bar, which is clamped onto the top of the rotor shaft using a chain. This fixture is aligned with the shaft axis by two parallel half-round brass location bars on the bottom of the fixture. Between the location bars is a hardened bushing that serves to guide a drill motor for machining the triggering feature.

Axial Keyphasor Installations

Axial Keyphasor installations are not recommended unless considerable care is taken in the evaluation of axial float, differential expansion, and probe gap setting. These installations should be avoided, as there have been many instances where the application failed to provide the required functionality of a repeatable once per event phase reference marker. A Keyphasor probe should never double for any other function such as Axial or Thrust position, Vibration or Eccentricity. However, a Keyphasor signal may be used to provide a Speed indication as well as a phase reference.

Accommodating Axial Movement

In the majority of cases, it will be possible to locate the Keyphasor triggering feature close to a thrust bearing, where axial movement and thermal growth are often minimal. In this instance, the design should be for a symmetrical trigger such as a round dimple or shallow hole.

However, because of machinery constraints, it may sometimes be necessary to locate the Keyphasor probe at a location where axial growth can be substantial. In such a case, an elongated triggering feature (slot) will be required to accommodate axial growth. Such a triggering notch or slot may also be required on machines having no thrust bearing, such as some electric motors, which depend on magnetic forces to keep the rotor approximately centered within the motor. With such designs, the coupling provides the axial stiffness to the driven unit.

Controlled Field Machining of Triggering Feature

In most cases, the machining of the Keyphasor trigger at an OEM or machinery repair or rebuild facility can be accomplished easily, since the rotor is accessible, and a multitude of milling tools is available. However, it can be riskier to machine a proper event trigger with the rotor in the machine.

For field machining, a Keyphasor triggering event can be produced using a ¾ in or 20 mm ball-nosed cutter. The nose cutter radius helps in preventing the formation of sharp internal corners that can be the source of stress risers. This size of cutter requires an in-situ machining tool or jig that is sometimes difficult to use in confined spaces.

Keyphasor Trigger Machining Jig

In many instances, a portable drilling jig – such as the one shown in Figures 6 & 7 – can be clamped onto the shaft, providing a safe and secure restraint for the cutter. The actual cutting of the triggering feature using such a jig takes only a few minutes.

Based upon specific machine design, speed and shaft diameter where the triggering feature is being placed, the size of the ball-nosed cutting head can potentially be reduced from ¾ in or 20 mm to as small as 8 mm to 10 mm diameter to accommodate more confined spaces.

Using the Jig

The jig is clamped onto the shaft using roller chain, and to prevent the shaft from damage, a soft protector strip is used (Figure 7). The machining fixture sits firmly on the two half-round brass location bars that accommodate installation on shafts with different diameters. In order to accommodate various shaft diameters, the chain is made up from suitable sections of different length – with a removable “master” connecting link that allows for easy changing of chain sections as needed.

Controlling Cutter Depth

The cutter machining depth can be reliably controlled by the use of a removable horseshoe-shaped setting washer of the required thickness. With this arrangement, the machining fixture is placed on the chosen section of shaft, and the cutter, washer and collar are loaded into the drill motor (Figure 8). When the cutter is in contact with the shaft, the setting collar is clamped into place on the cutter. The cutter is clamped into the drill chuck, and when the machining fixture is located and clamped into place, the cutter is inserted into an electric drill

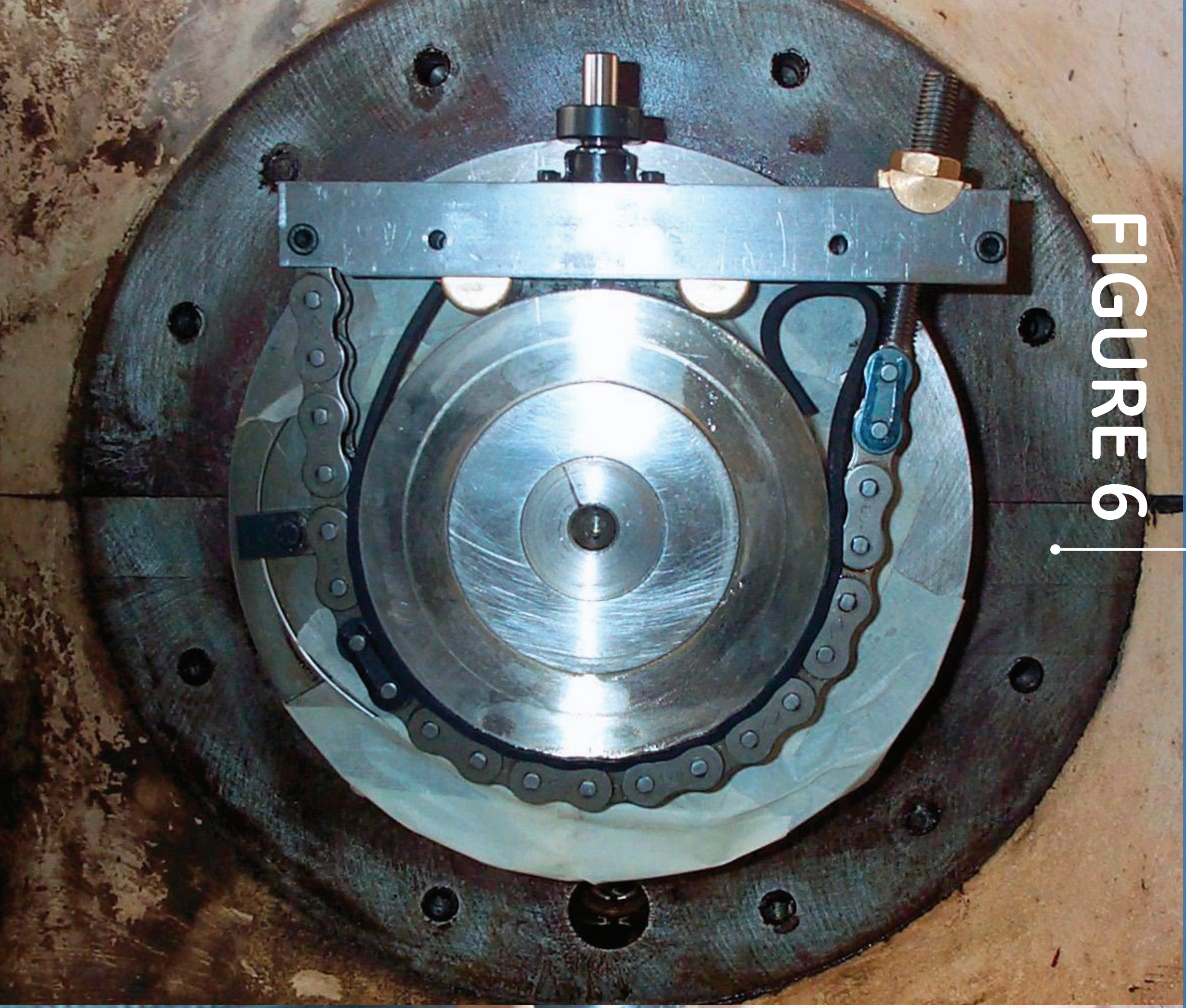


FIGURE 6

End-view photo of a drilling jig based on the design shown in Figure 5. In this example the Keyphasor jig is attached to the non-drive end of a gearbox shaft. The cutter is in place in the hardened guide bushing, but the drill motor has not yet been attached.

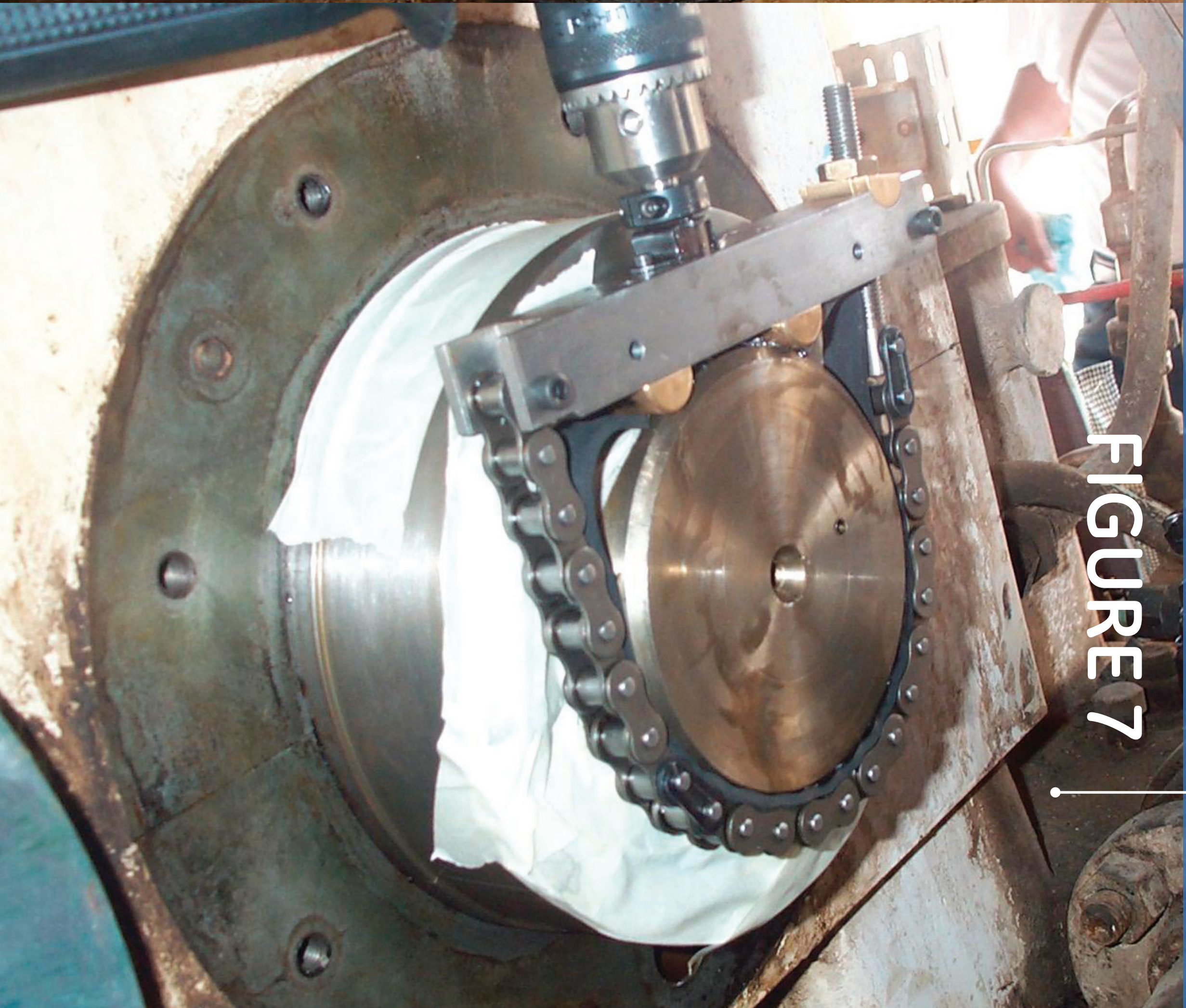


FIGURE 7

This photo shows a side view of the Keyphasor jig with the drill motor attached to the cutter. A rubber strip is in place to protect the shaft surface from being scratched by the chain links, and self-adhesive tape is used to prevent machining debris ingress into the journal bearing and shaft seals.

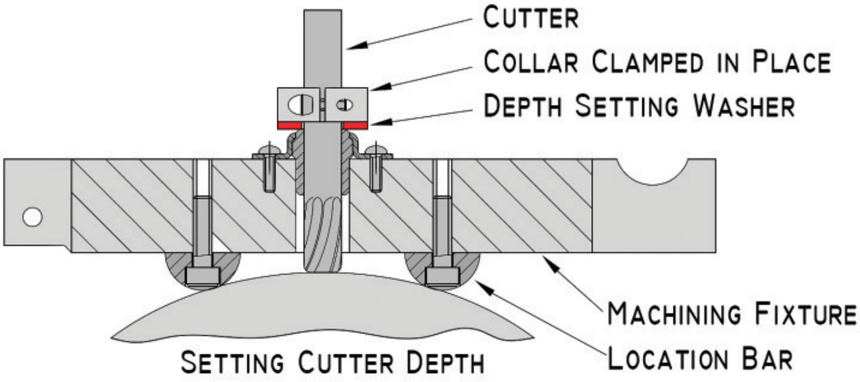

Cutter
Collar Clamped in Place
Depth Setting Washer
Machining Fixture
Location Bar
Setting Cutter Depth

FIGURE 8: End view of machining fixture showing clamped collar and removable depth-setting washer (red). In this example, the depth-setting washer was 2 mm thick.

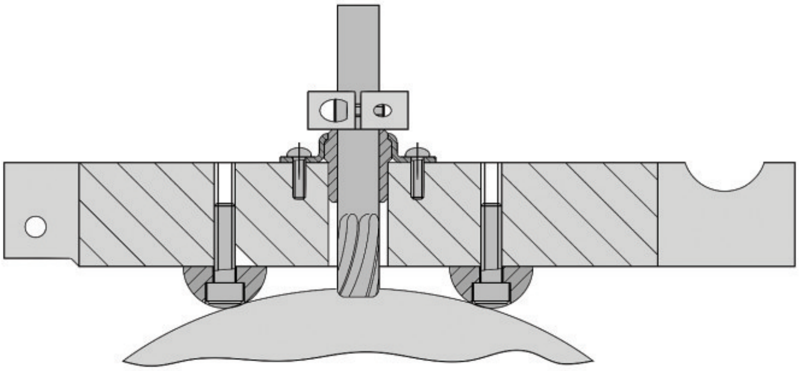


FIGURE 9: After removing the depth-setting washer, the cutting tool is allowed to cut to the depth of the washer thickness. In this example, the cutter can cut to a depth of 2 mm.

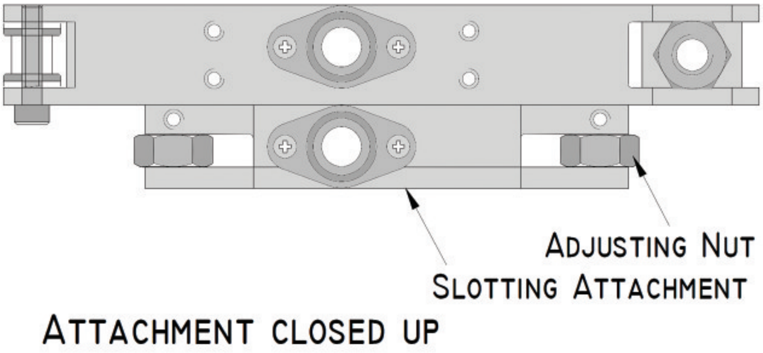

Adjusting Nut
Slotting Attachment
Attachment closed up

FIGURE 10: Top view of the machining fixture with elongation attachment installed. In this example, the attachment is closed up tightly against the main fixture.

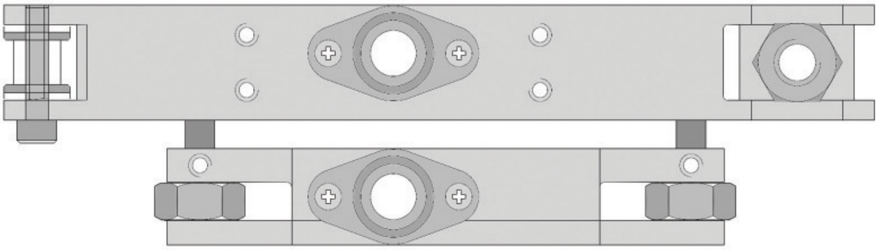


FIGURE 11: In this example, the elongation attachment is extended to its maximum distance away from the main fixture.

WITHOUT PHASE INFORMATION, OVERALL MACHINE CONDITION AND MACHINE FAULTS WOULD OFTEN BE VERY DIFFICULT, IF NOT IMPOSSIBLE, TO DIAGNOSE.

motor and the depth-setting washer is removed (Figure 9). This gives a very controlled and repeatable depth of cut based on the thickness of the setting washer..

The Elongation Attachment

The set-up illustrated so far caters for the machining of a circular triggering feature. To produce an elongated event (to accommodate axial movement of the shaft, relative to the Keyphasor probe), an additional attachment is required.

Attached to the basic jig is a similar machining fixture, with a similar electric drill motor support bushing. This additional fixture is carried on two studs that screw into the side of the basic jig (Figures 10 & 11). The studs used in this example permit a 10 mm wide triggering feature x 16 mm long to be formed. Longer studs can be used to machine longer triggering features.

The elongation attachment is fitted with two captive brass nuts to provide axial adjustment. The cutter is used in the same manner as in the basic jig. To produce the elongated event, the elongation attachment is simply moved slowly by half a turn of the nuts (as a pair) at a time – with the cutter effectively chain-drilling the slot. In spite of the multi-cut method of machining, the resultant event appears uniformly smooth.

Other Considerations

Various sizes of machining fixture can be built to accommodate shafts of different diameters. Also, machining fixtures can be built with the drill bushing offset, to allow positioning close to obstructions or changes in shaft section diameter. It should be noted that it is not good practice to locate triggering features close to changes in shaft diameters, near radial bearings or in high torque areas (load coupling area for example) of the shaft due to potential for increases in stress concentrations. ■

References

1. Orbit Vol.20 No.2, 1999. Forland, Clair, "Why phase information is important for diagnosing machinery problems."

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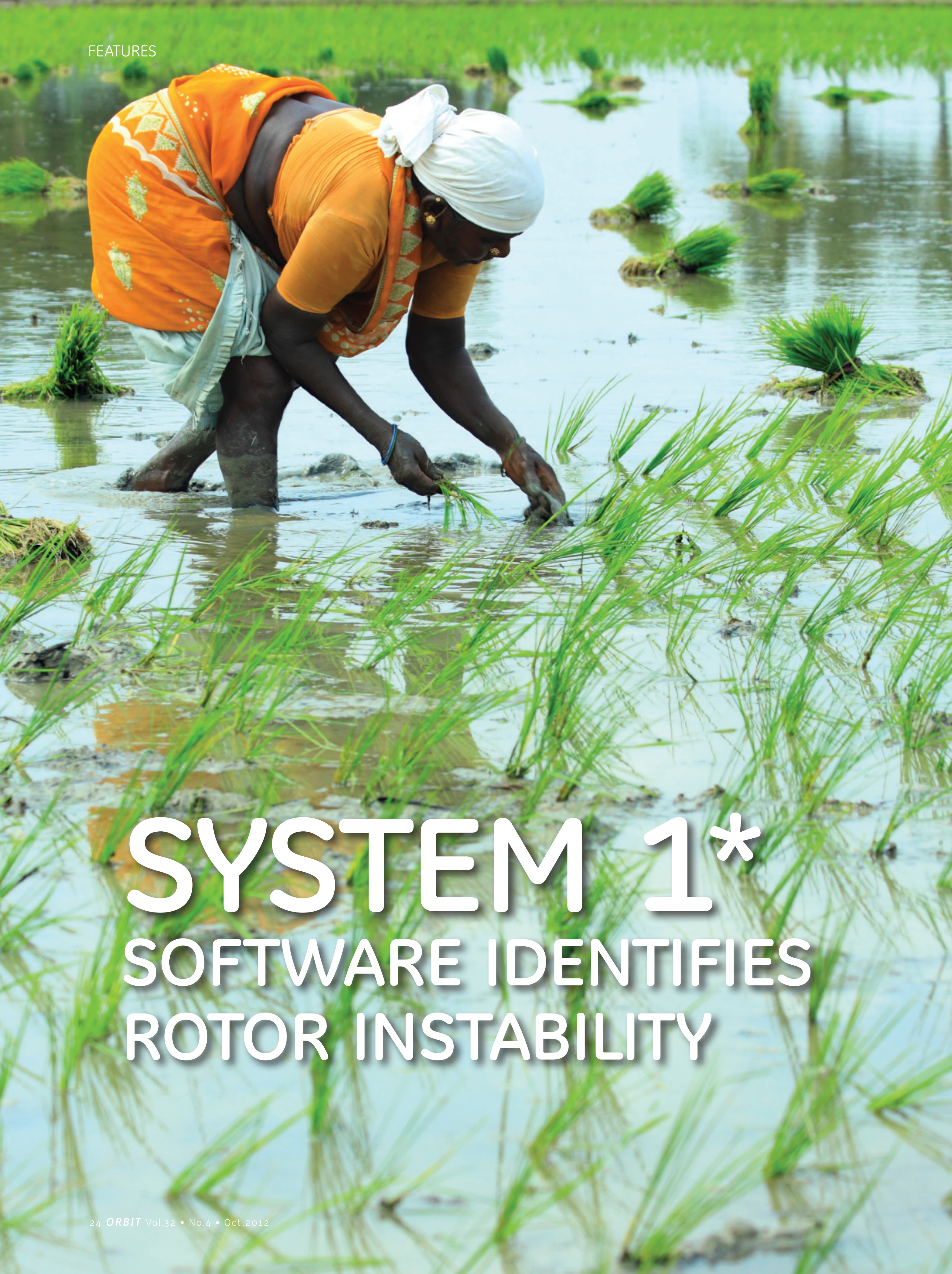

SYSTEM 1* SOFTWARE IDENTIFIES ROTOR INSTABILITY

THIS CASE STUDY DESCRIBES THE DIAGNOSTIC BENEFIT PROVIDED BY A SYSTEM 1 PLATFORM AT A LARGE FERTILIZER PLANT IN INDIA.

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Pankaj Sharma
Machinery Diagnostic Services Manager
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During the initial commissioning of a refurbished turbine-driven air compressor train, the machine tripped on high vibration measured at the steam turbine's Drive End (DE) bearing. The data was recorded by System1 software, and was later analyzed by a Machinery Diagnostics Services (MDS) engineer who determined that a fluid instability called "oil whip" had caused the high vibration trip. He produced a report for the OEM, recommending a modified bearing design to prevent recurrence of the problem. The manufacturer agreed with the engineer's recommendations, and modified the design of the turbine bearings based on his input. This event was an excellent example of how System 1 software is able to capture important diagnostic data during surprise plant upsets.

AERIAL VIEW OF THE DEEPAK FERTILIZER, TALOJA PLANT (NEAR MUMBAI, INDIA)

Historical Background

The events in this case history article occurred at Deepak Fertilizer and Petrochemicals Company Ltd.'s fertilizer plant in Taloja, India, during December of 2010. This plant was originally built in 1982, and has an output of about 1000 tons per year of Ammonium Nitrate. As part of a plant-wide upgrade project, the main air compressor in the weak nitric acid plant was overhauled using refurbished parts. During the air compressor overhaul, a Bently Nevada 3500 Series monitoring system was installed, along with System1 software.

The air compressor described in this article had been installed for more than 25 years at the time of its overhaul. The machine train includes a steam turbine, two separate centrifugal air compressors, and a turbo-expander (Figure 1). The expander reduces the required steam turbine power output during times when reactor off-gas is available to provide power via the expander.

Asset Criticality Note: The production of nitric acid is a necessary step in the manufacture of fertilizer. The high-volume air compressor is highly critical to the operation of the weak nitric acid plant, due to its status as a single (unspared) machine, the high capital investment that it represents, and the significant cost of production down time that would result from an air compressor failure.

Sequence of Events

During the compressor startup on December 8, 2010 at 5 PM, the steam turbine tripped on high vibration at the Drive End (DE) bearing. The observed symptoms are summarized below (Figure 2).

- Vibration amplitude on the turbine DE bearing (6104-4A2 & 4B2) suddenly increased.
- No changes in process conditions or bearing temperature were observed.
- Possible steam pressure changes were suspected, but could not be confirmed.
- A similar rise in vibration amplitude had been observed on November 27, 2010. However, the steam turbine did not trip during that occurrence.

The local Bently Nevada MDS Engineer was conducting a training program at the time of the trip, so he was not immediately available to review the data. However, after completing his training commitments, he was able to evaluate the data and come to an understanding of the root cause of the vibration rise. This was an excellent example of how System 1 software can capture high-resolution data whenever a surprise plant upset occurs, and facilitate diagnostic analysis of this data as soon as someone is available to review it.

Vibration Data Plots and Analysis

The System 1 Event Manager (Figure 2) showed that the steam turbine tripped on high vibration amplitude at the DE Bearing. However, further investigation revealed that the vibration actually began to increase almost one hour before the trip finally occurred.

As shown in Figure 3, the rise in "Direct" (unfiltered) amplitude was not caused by an increase in 1X amplitudes.

When observing the data that was collected around the time of the trip (Figure 4), it appears that the vibration amplitudes exceeded the "Danger" (automatic trip) setpoint at about 16:55:57 hours, but that vibration then continued to increase for about 9 more seconds (to about 16:56:06), before finally falling below the Danger setpoint at about 16:56:08.

This scenario may lead us to believe that, after the vibration amplitude exceeded the Danger setpoint, there was an unusual delay with the trip relay that prevented a reduction in vibration until after 16:56:06. However, the actual trip delay was verified to be only 3 seconds, which is normal. What the data really shows is a somewhat unexpected instance of vibration continuing to increase for a few seconds after the turbine trip.

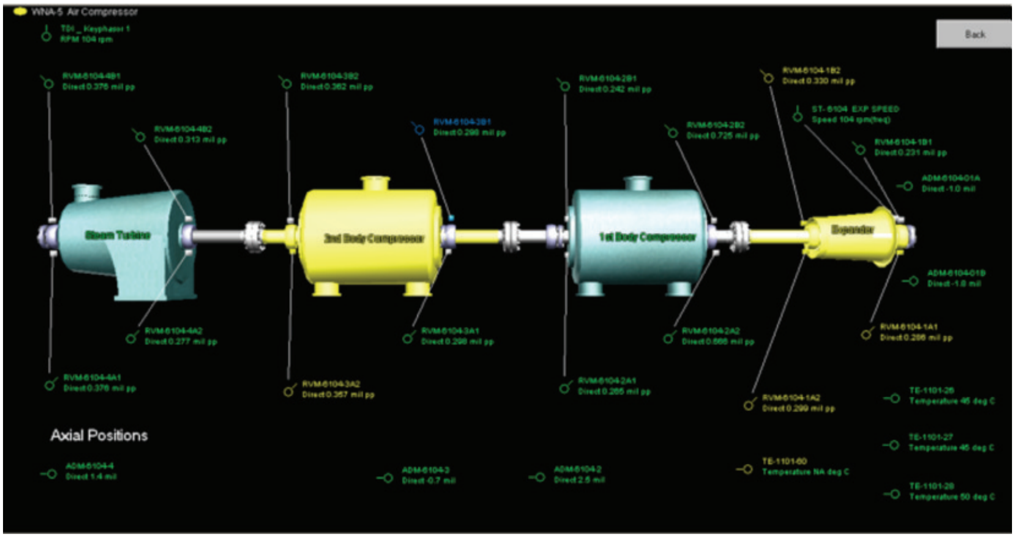
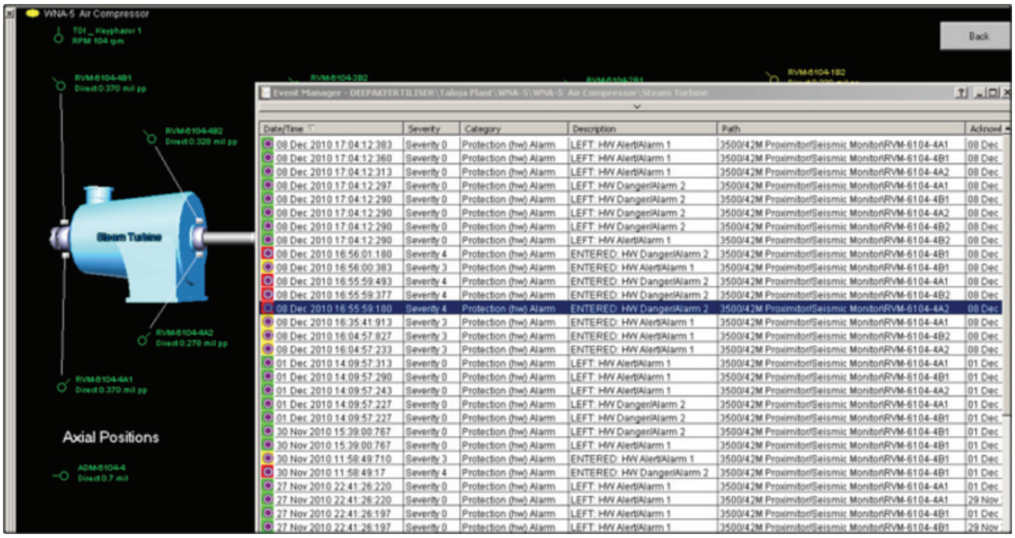

WNA-5 Air Compressor
TDI _ Keyphasor 1
RPM 104 rpm
Back
RVM-6104-4B1
Direct 0.376 mil pp
RVM-6104-3B2
Direct 0.362 mil pp
RVM-6104-2B1
Direct 0.242 mil pp
RVM-6104-1B2
Direct 0.330 mil pp
ST- 6104 EXP SPEED
Speed 104 rpm(freq)
RVM-6104-4B2
Direct 0.313 mil pp
RVM-6104-3B1
Direct 0.298 mil pp
RVM-6104-2B2
Direct 0.725 mil pp
RVM-6104-1B1
Direct 0.231 mil pp
ADM-610401A
Direct -1.0 mil
Steam Turbine
2nd Body Compressor
1st Body Compressor
Expander
ADM-610401B
Direct -1.8 mil
RVM-6104-4A2
Direct 0.277 mil pp
RVM-6104-3A1
Direct 0.298 mil pp
RVM-6104-2A2
Direct 0.666 mil pp
RVM-6104-1A1
Direct 0.286 mil pp
RVM-6104-4A1
Direct 0.376 mil pp
RVM-6104-3A2
Direct 0.357 mil pp
RVM-6104-2A1
Direct 0.265 mil pp
RVM-6104-1A2
Direct 0.299 mil pp
TE-1101-26
Temperature 45 deg C
Axial Positions
TE-1101-27
Temperature 45 deg C
TE-1101-60
Temperature NA deg C
ADM-6104-4
Direct 1.4 mil
ADM-6104-3
Direct -0.7 mil
ADM-6104-2
Direct 2.5 mil
TE-1101-28
Temperature 50 deg C

FIGURE 1: System 1 Machine Train Diagram for the air compressor. From left to right are the Steam Turbine, Second Body Compressor, First Body Compressor, and Expander. Current values for monitored parameters are shown in the text blocks.


WNA-5 Air Compressor
TDI _ Keyphasor 1
RPM 104 rpm
Back
RVM-6104-4B1
Direct 0.370 mil pp
RVM-6104-4B2
Direct 0.328 mil pp
Steam Turbine
RVM-6104-4A2
Direct 0.278 mil pp
RVM-6104-4A1
Direct 0.370 mil pp
Axial Positions
ADM-6104-4
Direct 0.7 mil
RVM-6104-3B2
RVM-6104-2B1
RVM-6104-1B2

Event Manager - DEEPAKFERTILISER\Taloja Plant\WNA-5\WNA-5 Air Compressor\Steam Turbine

Date/Time	Severity	Category	Description	Path	Acknowl
08 Dec 2010 17:04:12:383	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	08 Dec
08 Dec 2010 17:04:12:360	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	08 Dec
08 Dec 2010 17:04:12:313	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A2	08 Dec
08 Dec 2010 17:04:12:297	Severity 0	Protection (hw) Alarm	LEFT: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	08 Dec
08 Dec 2010 17:04:12:290	Severity 0	Protection (hw) Alarm	LEFT: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	08 Dec
08 Dec 2010 17:04:12:290	Severity 0	Protection (hw) Alarm	LEFT: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A2	08 Dec
08 Dec 2010 17:04:12:290	Severity 0	Protection (hw) Alarm	LEFT: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B2	08 Dec
08 Dec 2010 17:04:12:290	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B2	08 Dec
08 Dec 2010 16:56:01:180	Severity 4	Protection (hw) Alarm	ENTERED: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	08 Dec
08 Dec 2010 16:56:00:383	Severity 3	Protection (hw) Alarm	ENTERED: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	08 Dec
08 Dec 2010 16:55:59:493	Severity 4	Protection (hw) Alarm	ENTERED: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	08 Dec
08 Dec 2010 16:55:59:377	Severity 4	Protection (hw) Alarm	ENTERED: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B2	08 Dec
08 Dec 2010 16:55:59:180	Severity 4	Protection (hw) Alarm	ENTERED: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A2	08 Dec
08 Dec 2010 16:35:41:913	Severity 3	Protection (hw) Alarm	ENTERED: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	08 Dec
08 Dec 2010 16:04:57:827	Severity 3	Protection (hw) Alarm	ENTERED: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B2	08 Dec
08 Dec 2010 16:04:57:233	Severity 3	Protection (hw) Alarm	ENTERED: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A2	08 Dec
01 Dec 2010 14:09:57:313	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	01 Dec
01 Dec 2010 14:09:57:290	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	01 Dec
01 Dec 2010 14:09:57:243	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A2	01 Dec
01 Dec 2010 14:09:57:227	Severity 0	Protection (hw) Alarm	LEFT: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	01 Dec
01 Dec 2010 14:09:57:227	Severity 0	Protection (hw) Alarm	LEFT: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	01 Dec
30 Nov 2010 15:39:00:767	Severity 0	Protection (hw) Alarm	LEFT: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	01 Dec
30 Nov 2010 15:39:00:767	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	01 Dec
30 Nov 2010 11:58:49:710	Severity 3	Protection (hw) Alarm	ENTERED: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	01 Dec
30 Nov 2010 11:58:49:17	Severity 4	Protection (hw) Alarm	ENTERED: HW Danger/Alarm 2	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	01 Dec
27 Nov 2010 22:41:26:220	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	01 Dec
27 Nov 2010 22:41:26:220	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4A1	29 Nov
27 Nov 2010 22:41:26:197	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	01 Dec
27 Nov 2010 22:41:26:197	Severity 0	Protection (hw) Alarm	LEFT: HW Alert/Alarm 1	3500/42M Proximitor/Seismic Monitor/RVM-6104-4B1	29 Nov

FIGURE 2: System 1 Event Manager showing steam turbine trip at 16:55:59:180 Hours on December 8, 2010.

The reason for this behavior can be understood from the trend of turbine speed as shown in Figure 5. At 16:56:01 hours, a small dip in speed was observed and afterward, the speed ramped up from 6950 RPM to 7436 RPM at 16:56:04. This indicates that even though the steam turbine had tripped, there was some other source of energy available to the rotor which caused its speed to for ramp up even after the steam turbine tripped. This energy most likely came from the Expander which had just come into operation after furnace ignition.

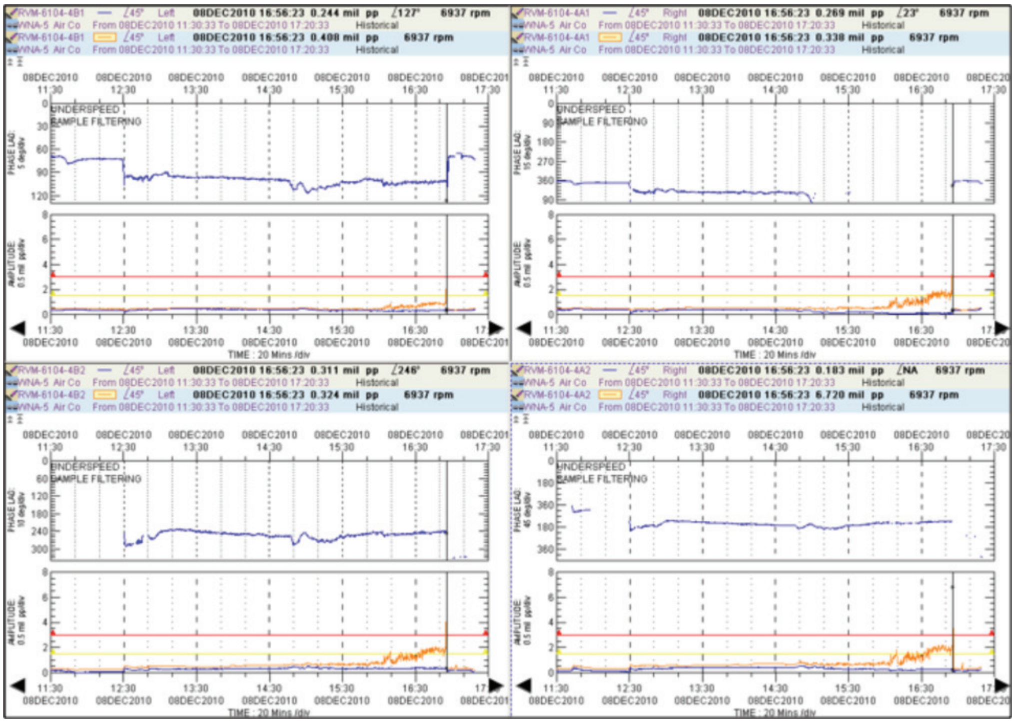


FIGURE 3: Direct Vibration levels began increasing about an hour before the trip (orange curve in lower half of plots) but no changes were observed in 1X filtered amplitudes (blue curves in upper half of plots). This meant that the increase in amplitude was due to increasing vibration levels at NOT 1X frequencies.

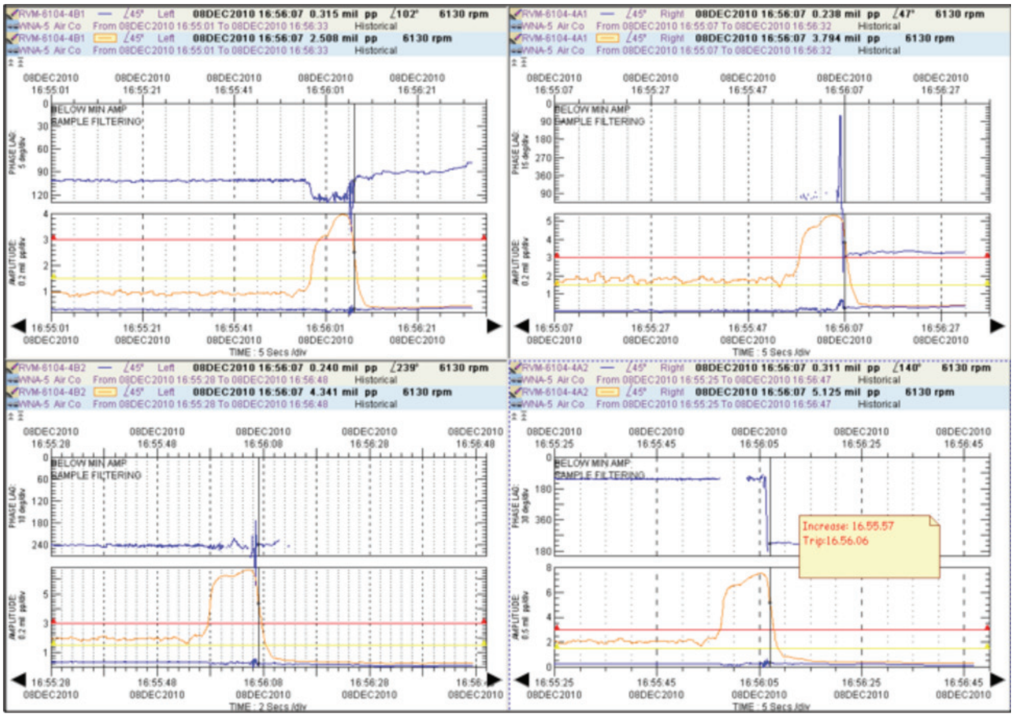


FIGURE 4: Plots have been rescaled to look very closely at the time just before and after the steam turbine trip.

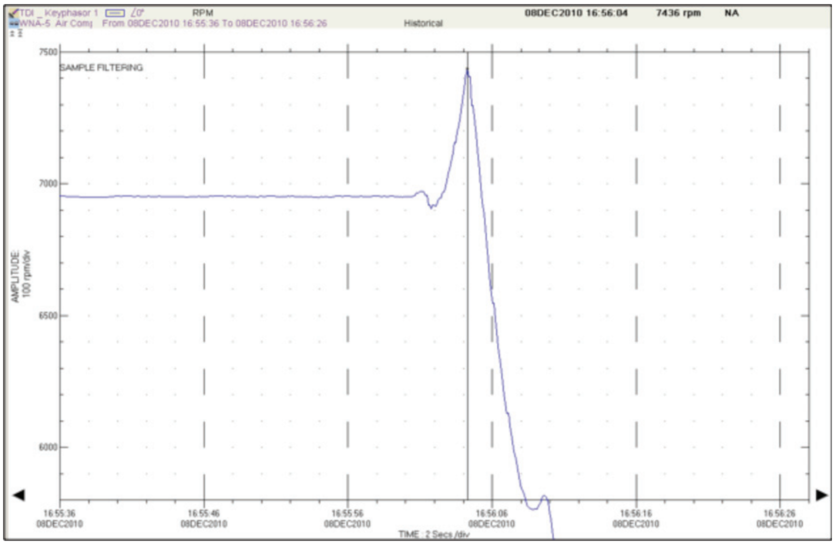


FIGURE 5: The steam turbine Keyphasor* signal shows that machine speed ramped up from 6950 to 7436 rpm before dropping off during the machine coast-down.

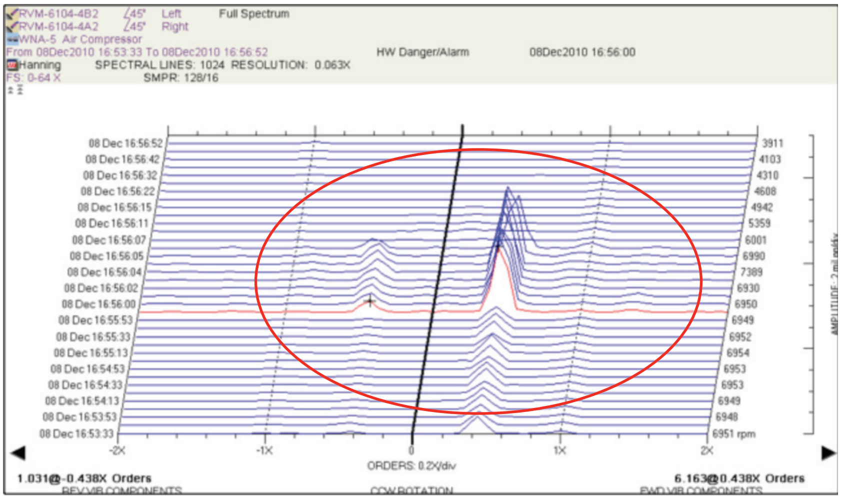


FIGURE 6: Subsynchronous 0.438X frequency that was responsible for increased Direct vibration amplitudes.

Study of the full spectrum waterfall plots (Figure 6) clearly indicates elevated amplitudes of forward 0.438X frequency (with smaller reverse components). This vibration was present with lower amplitudes earlier and was responsible for gradually increasing overall amplitudes prior to 1 hour before the actual trip.

As the turbine speed increased following its trip, the subsynchronous force was not eliminated – causing turbine DE bearing vibration to increase until 16.56.06 hours.

A review of rotor radial position from the Average Shaft Centerline Plots (Figure 7) showed some very interesting information. Just before the steam turbine tripped on high vibration amplitude, its rotor started moving towards the bearing center (reduced eccentricity ratio). This position change correlated with the timing of the increased high vibration amplitude.

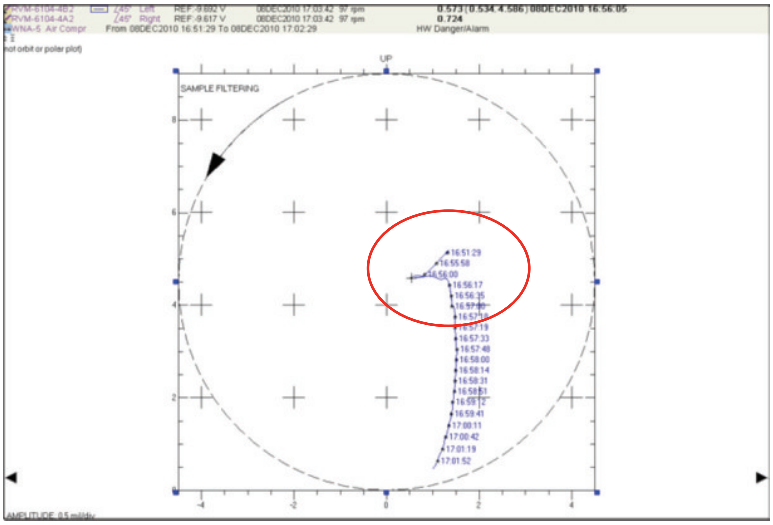


FIGURE 7: Average position of the steam turbine rotor appears to be moving towards a lower eccentricity ratio (closer to the centerline of the bearing bore) at the time of the elevated vibration that led to the trip. From the time of the trip onwards, the shaft centerline position slowly drifts down to the bottom of the bearing clearance – which is a very normal response.

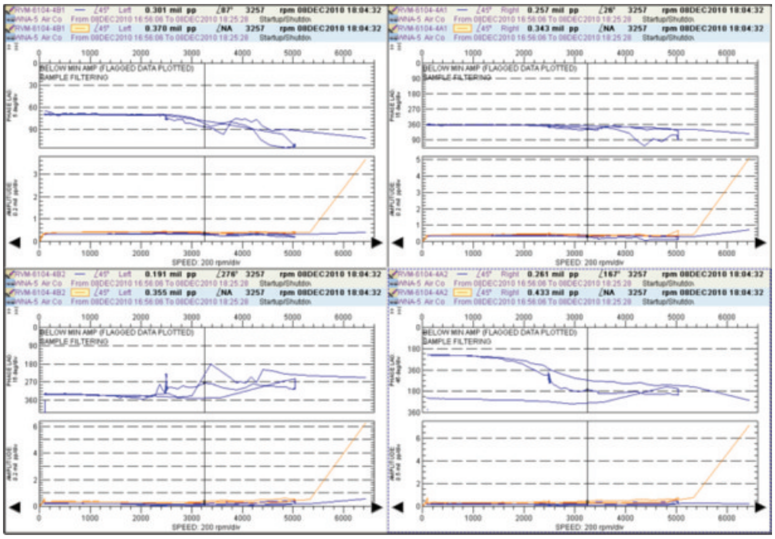


FIGURE 8: Bode plots indicated a natural frequency near 3200 cpm (~53 Hz).

Bode plots (Figure 8) indicated hints of a rotor balance resonance near 3200 cpm (53Hz), as observed from phase changes. The subsynchronous frequency component that was excited during the vibration rise is also near 53Hz. These are starting to look like classic symptoms of a fluid-induced rotor instability.

Analysis of various other plots (Figures 9 through 16) verified oil whip as the cause for the observed vibration rise at the turbine DE bearing. Because of the excess power that was being supplied by the expander, the steam turbine was unloaded, causing the rotor to move towards the bearing center. Because the steam turbine bearings are fluid-film sleeve bearings, the rotor operating at a lower eccentricity ratio (closer to

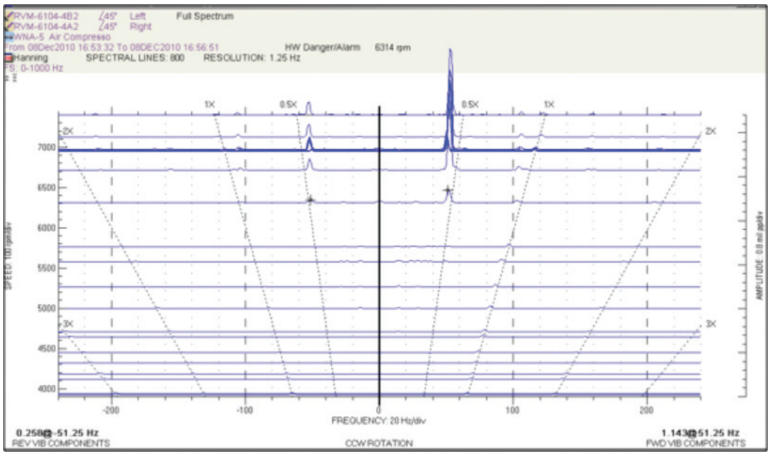


FIGURE 9: Cascade plots for turbine trip and overspeed event. The rotor instability appeared above 6314 RPM at a small amplitude. Then, when the expander came online and unloaded the steam turbine, the instability suddenly became excited at a much higher amplitude. This is also shown in the waterfall plot in Figure 6.

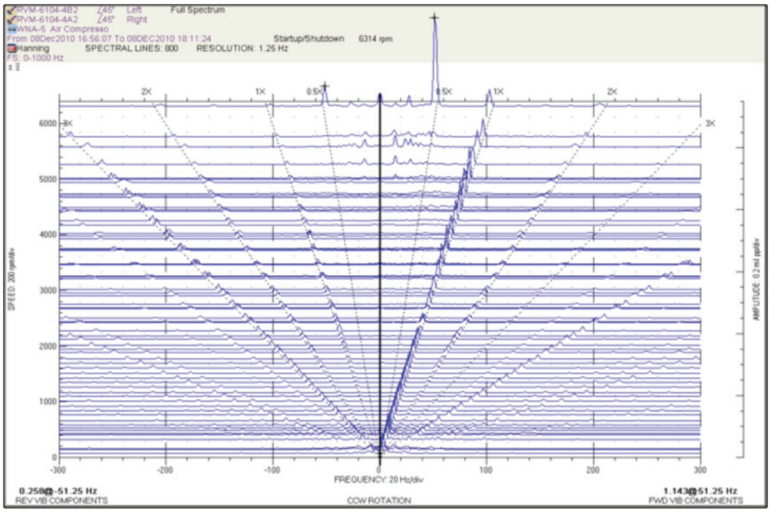


FIGURE 10: The cascade plot for the machine coast-down shows that the subsynchronous vibration components rapidly dissipated as the machine ramped down. It was essentially gone by the time the machine speed slowed to 4000 rpm.

the center of the bearing clearance) will be susceptible to the excitation of natural frequencies due to a higher than normal oil wedge tangential force.

One characteristic of oil whip is that it appears when exceeding a specific speed of operation, which is referred as "Threshold of stability". For this particular event, the threshold of stability was approximately 6314 RPM. Generally, the designer's intention is to always keep this threshold of stability above the normal operating speed of the rotor, so that the rotor does not experience the instability during normal operation. However, due to viscosity changes, increased clearances, certain alignment conditions, etc., the threshold of stability can sometimes shift to a significantly lower speed, causing rotor sub synchronous vibration to occur at normal operating speed. This is exactly what happened in this case.

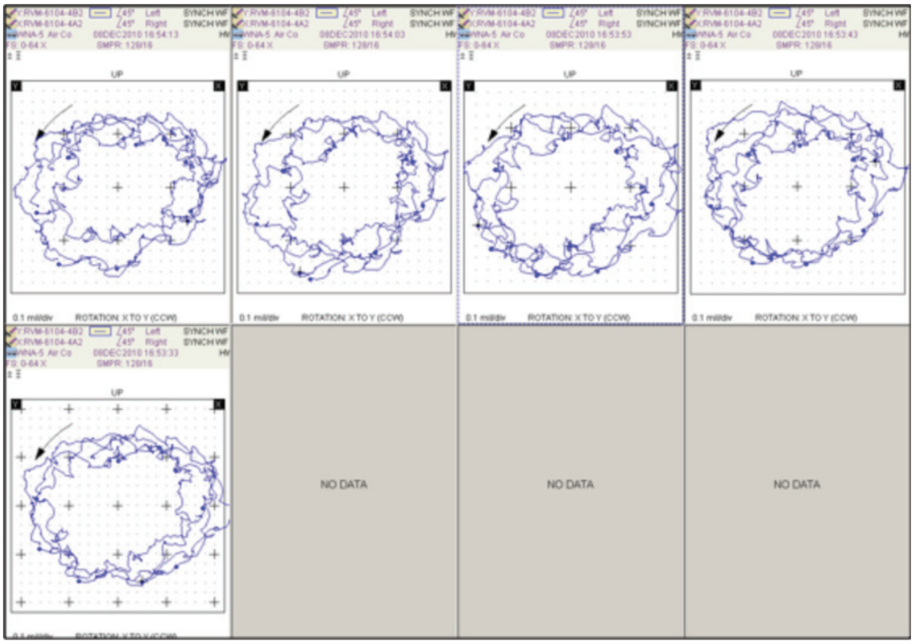

UP
0.1 mil/div
ROTATION: X TO Y (CCW)
NO DATA
NO DATA
NO DATA

FIGURE 11: These orbit plots show the erratic shaft precession that existed as the whip condition was becoming established (from 16:53:33 to 16:54:13), just a few minutes before the high vibration trip occurred. The data for these plots was collected at 10 second intervals, starting with the plot in the lower row, and then moving from right to left through the plots in the upper row.

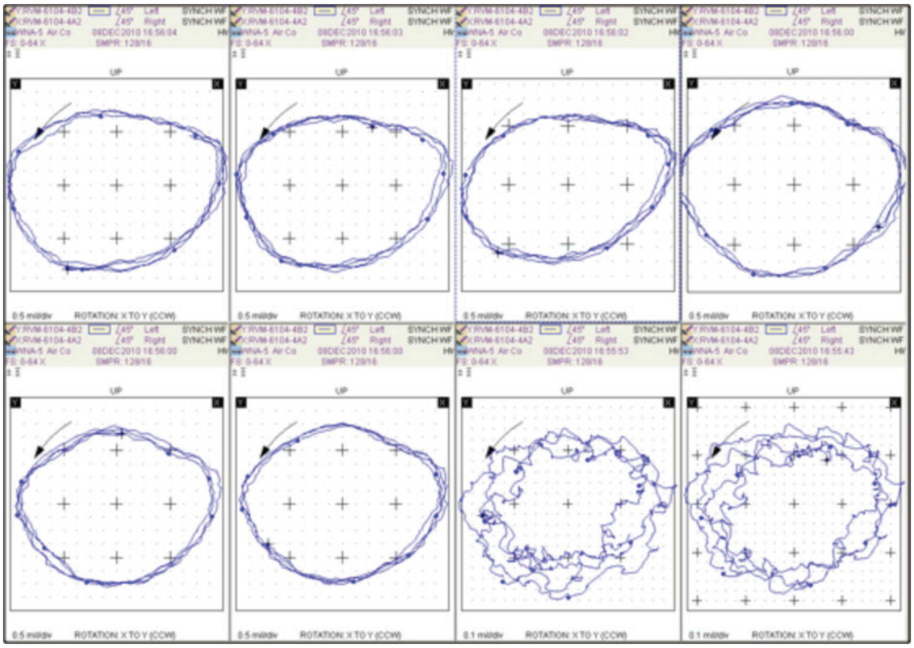

UP
0.5 mil/div
0.1 mil/div
ROTATION: X TO Y (CCW)

FIGURE 12: These orbit plots show the oil whip symptoms becoming more pronounced from the chaotic motion at 16:55:43 in the lower right corner, through a very circular pattern starting at 16:56:00 (second plot from left in lower row).

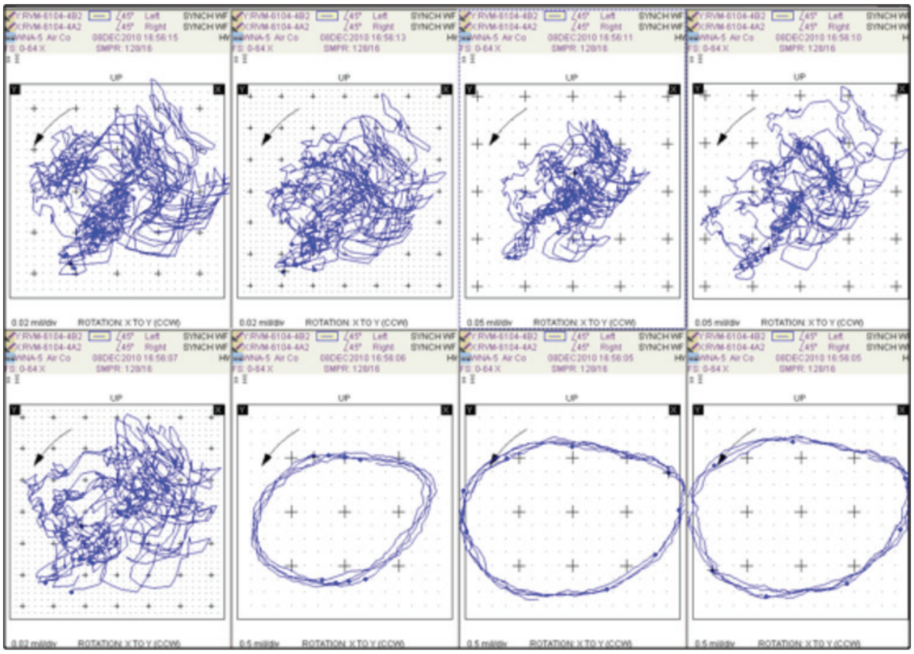


FIGURE 13: These orbit plots show how the shaft centerline movement changed from being very circular during the full whip condition at 16:56:05 (lower right plot), through a period of chaotic movement as the machine coasted down following the trip.

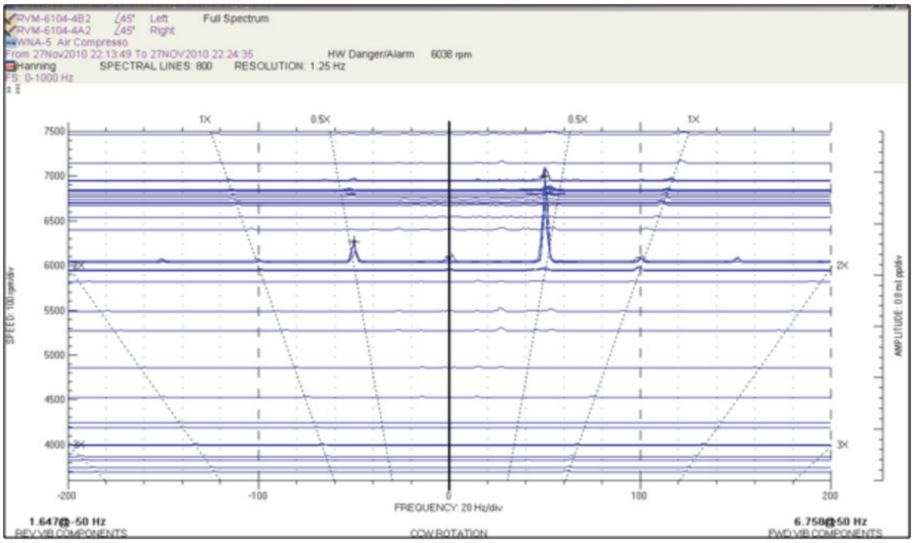


FIGURE 14: For the 27NOV2010 event, the excitation frequency was also about 50 Hz. Observe that a strong forward precession first appeared when reaching 6038 rpm as machine speed increased. **NOTE:** Interestingly, a similar high vibration & oil whip event was recorded by the System 1 installation about a week and a half earlier (November 27, 2010) as shown in Figure 15. However, for this earlier occurrence, the threshold of stability was closer to 6038 rpm. The effects disappeared and then reappeared after passing through 7000 rpm.

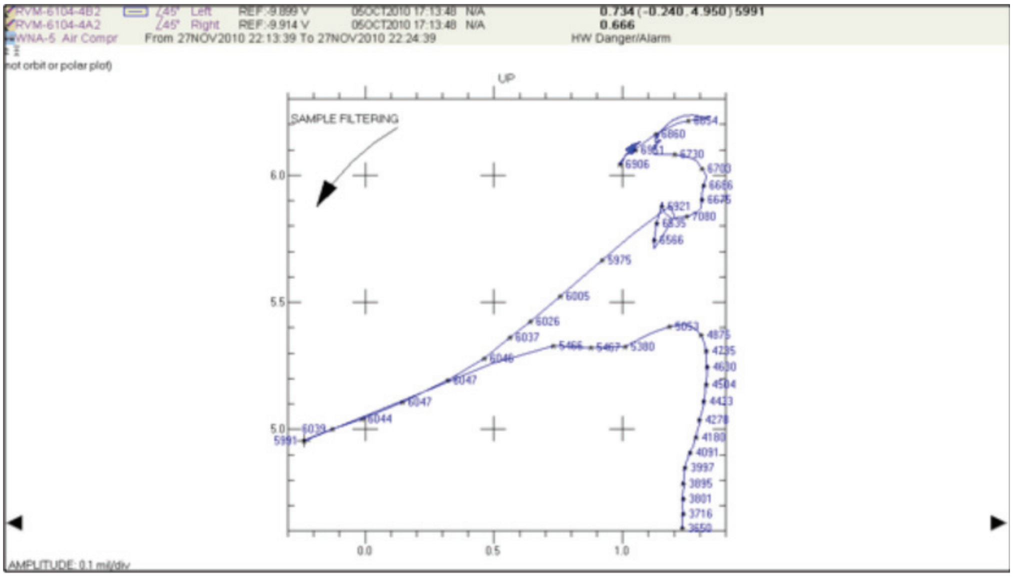


FIGURE 15: Average shaft centerline plot for 27NOV2010, showing the shaft approaching the bearing centerline as the steam turbine became unloaded when the expander came online. Observe the similarity of this plot to Figures 7 and 16, which show the 08DEC2010 event.

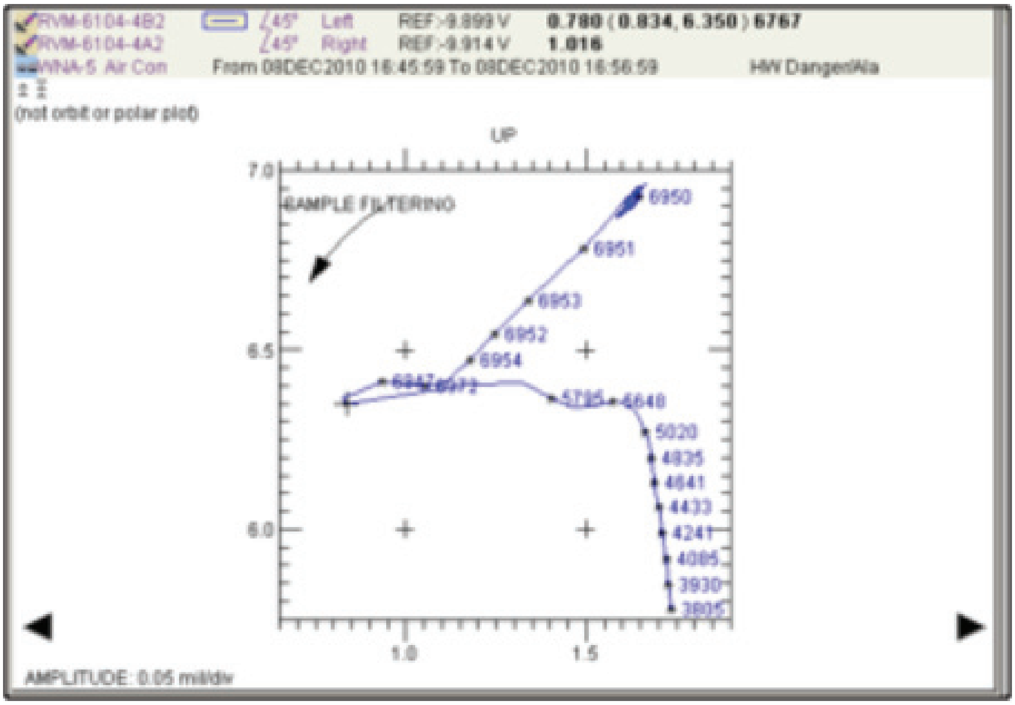


FIGURE 16: Observe the behavioral similarity in average shaft centerline plots between this example from 08DEC2010, and Figure 15, from 27NOV2010. In both cases, the steam turbine rotor moved towards the centerline of its bearing.

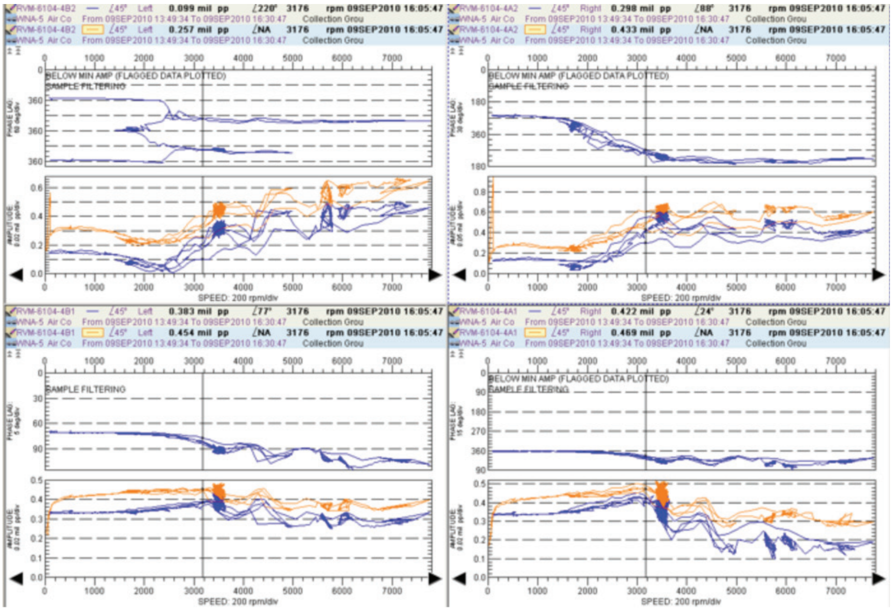


FIGURE 17: Turbine solo run from routine overspeed test of 09SEP2010, showing small amplitudes of vibration.

Additional Information

Figure 17 shows Bode plots of steam turbine vibration recorded during a solo run of the turbine without being connected to the compressor and expander connected. No whip was observed. This suggests that the oil whip events described above occurred when the turbine was unloaded after the Expander came online during loading after the plant startup.

Analysis Summary

From the vibration data recorded by the System 1 platform for both the high vibration events of 27NOV2010 and 08DEC2010, a strong behavioral correlation of the rotor moving to a lower eccentricity ratio was observed. When a rotor is in such a position, it is much more susceptible to oil whip, which is the subsynchronous excitation of a natural rotor frequency caused by the tangential component of the lubricating oil wedge.

When comparing the two events described here, the threshold of stability was not the same for the two events. This suggests that the observed oil whip conditions were strongly dependent on the effects of the expander loading on the steam turbine. Further, instability symptoms were not observed during the solo run test of the steam turbine.

Unloading of the steam turbine rotor during expander operation has been identified as the most likely cause for the observed intermittent oil whip behavior. Process conditions were critically reviewed by the plant technology team to verify that the loading sequence had been implemented correctly, without errors or unexpected mismatches.

Vibration data clearly indicated the need for changes in bearing design to push the threshold of stability beyond normal machine operating speed so future instances of oil whip can be avoided. These vibration observations and conclusions were submitted to OEM on 09DEC2010. The OEM agreed to the observations and suggested modifications to modify the design of the steam turbine's bearings.

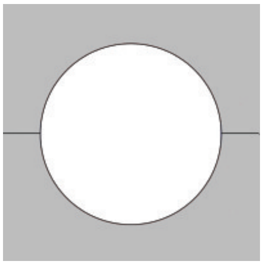


FIGURE 18A: Original cylindrical bore of journal bearing housing.

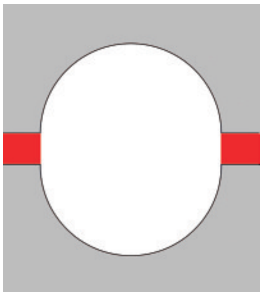


FIGURE 18B: Spacers (red) inserted between housing halves for machining (greatly exaggerated for illustration).

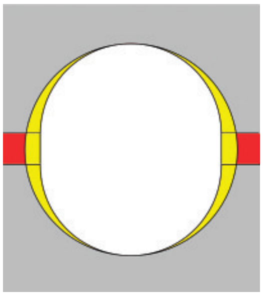


FIGURE 18C: Metal to be removed (yellow) by machining the bore to a larger diameter.

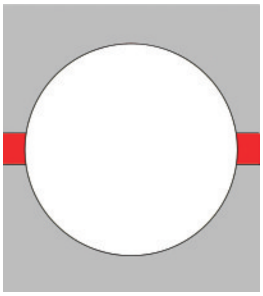


FIGURE 18D: Bearing housing after re-boring to larger diameter.

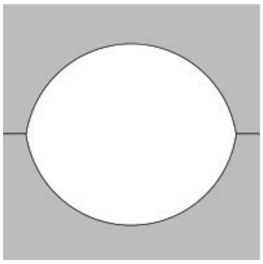


FIGURE 18E: After removing the spacer plates, the vertical dimension of the bearing is unchanged from its original value. However, the horizontal dimension has been increased, giving the bore its characteristic "lemon" profile.

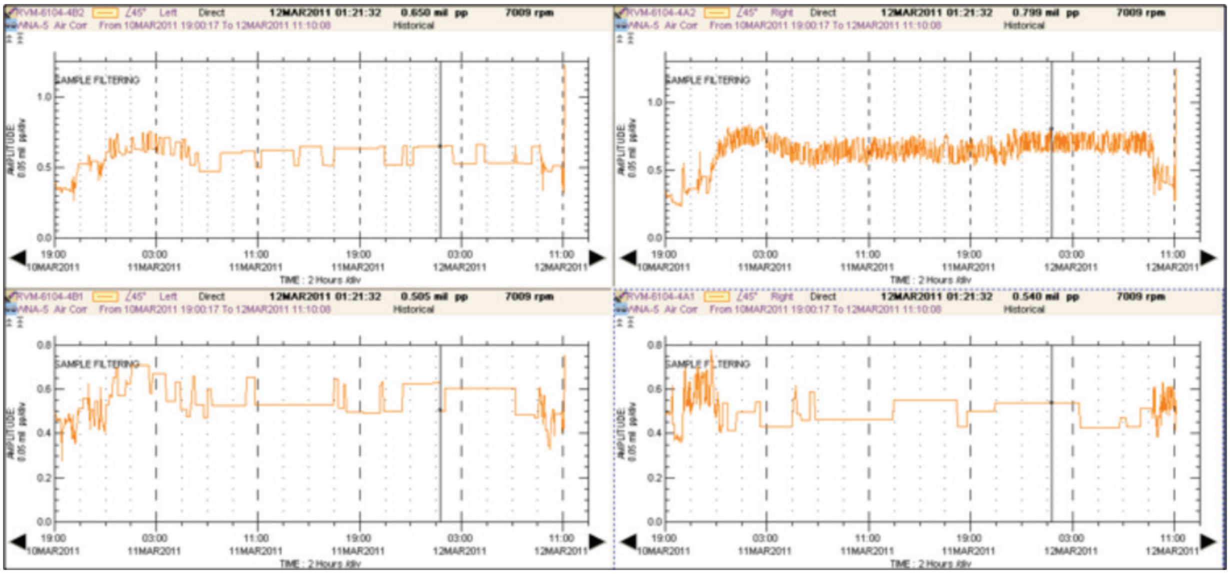


FIGURE 19: Vibration amplitude trends remained within acceptable limits after the bearing design changes were implemented.

Bearing Design Modifications

As shown in Figure 18A through 18E, the steam turbine DE (rear) bearing was machined to give 0.45 mm of ellipticity in its horizontal axis. With the new "lemon bore" design, the threshold of stability could successfully be pushed beyond an operating speed of 7000 rpm. Also, the excess lube oil flow to the steam turbine NDE (front) bearing was reduced by modifying the oil supply hole in the bearing housing.

After Bearing Modification

As shown in Figure 19, Direct vibration amplitudes trended at normal amplitudes following the installation of the modified bearings. With the bearing design changes, the air compressor has run normally without any recurrence of the troublesome fluid instability.

Summary

The installation and commissioning of the System 1 platform allowed us to capture the vital vibration data that was measured during unanticipated periods of rotor instability. The availability of this historical information allowed us to analyze the data and arrive at the root cause of the unexpected vibration rise. The System 1 data allowed us to perform effective vibration analysis and provide convincing evidence to the OEM for the necessity of bearing design changes. In the truest sense, the System 1 installation was ready to capture any surprise events for changes in machine behavior. ■

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THE HIGH-VOLUME AIR COMPRESSOR IS HIGHLY CRITICAL TO THE OPERATION OF THE WEAK NITRIC ACID PLANT, DUE TO ITS STATUS AS A SINGLE (UNSPARED) MACHINE, THE HIGH CAPITAL INVESTMENT THAT IT REPRESENTS, AND THE SIGNIFICANT COST OF PRODUCTION DOWN TIME THAT WOULD RESULT FROM AN AIR COMPRESSOR FAILURE.

DIAGNOSING GENERATOR ROTOR THERMAL SENSITIVITY

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A thermally-sensitive rotor can cause excessive generator vibration amplitudes at normal operating conditions when field current is raised to increase output voltage (and associated MVAR loading). In most cases, it will not preclude generator operation. However, it may limit operation at high field current, or lead to reduced generation capacity and efficiency. It can also negatively impact long-term operation due to repeated forced outages.

There are many possible root causes of this problem, including blocked ventilation and cooling media, shorted turns, coil insulation degradation due to moisture ingression, and malfunction of the cooling gas system. Any of these factors that produce a non-uniform circumferential thermal distribution around the rotor will cause asymmetric thermal expansion and rotor bow, resulting in an increase in vibration response (Reference 1).

Appropriate condition monitoring and diagnostic tools are required to pinpoint the exact root cause such that the appropriate remedial action can be taken. Experience shows that most generator rotors have at least some shorted turns – especially if they have been in service for a long time. This situation is considered to be acceptable unless it causes excessive vibration amplitude at normal operating conditions.

This article demonstrates the use of a modern online machine condition monitoring system to identify a thermally sensitive rotor in a 100 MW hydrogen-cooled generator unit installed in a combined cycle power plant. Other technologies (magnetic flux probe testing, visual inspection and electrical testing) were used to confirm the diagnosis, leading to the decision to repair the rotor windings.

The Site

EGAT's Bangpakong site is the largest power generating facility in Thailand (Reference 2), accounting for more than 25% of country's electricity supply. The plant is located about 100 km from Bangkok in a major industrial area in the eastern part of the country. The complex includes four thermal plant blocks and five combined cycle plant blocks with a total installed capacity of 3,680 MW. The subject unit is a 100 MW, 3000 rpm, hydrogen-cooled generator driven by a General Electric (GE) Frame 9E heavy duty gas turbine, installed in one of the combined cycle blocks.

Event History

The generator discussed in this case study has been in operation since 1992. It was not initially equipped with eddy current displacement transducers for measuring shaft vibration. However, "seismic" vibration transducers (moving-coil velocity sensors) were installed on the bearing caps of both the gas turbine and generator. The vibration data was monitored directly by the machine control system for machine protection and operation.

Due to a significant non-recoverable degradation in plant efficiency, Bangpakong's management team planned a major refurbishment to the gas turbine units of combined cycle blocks 3 and 4 in 2007. Project scope included the installation of machine condition and performance monitoring systems, with the following features:

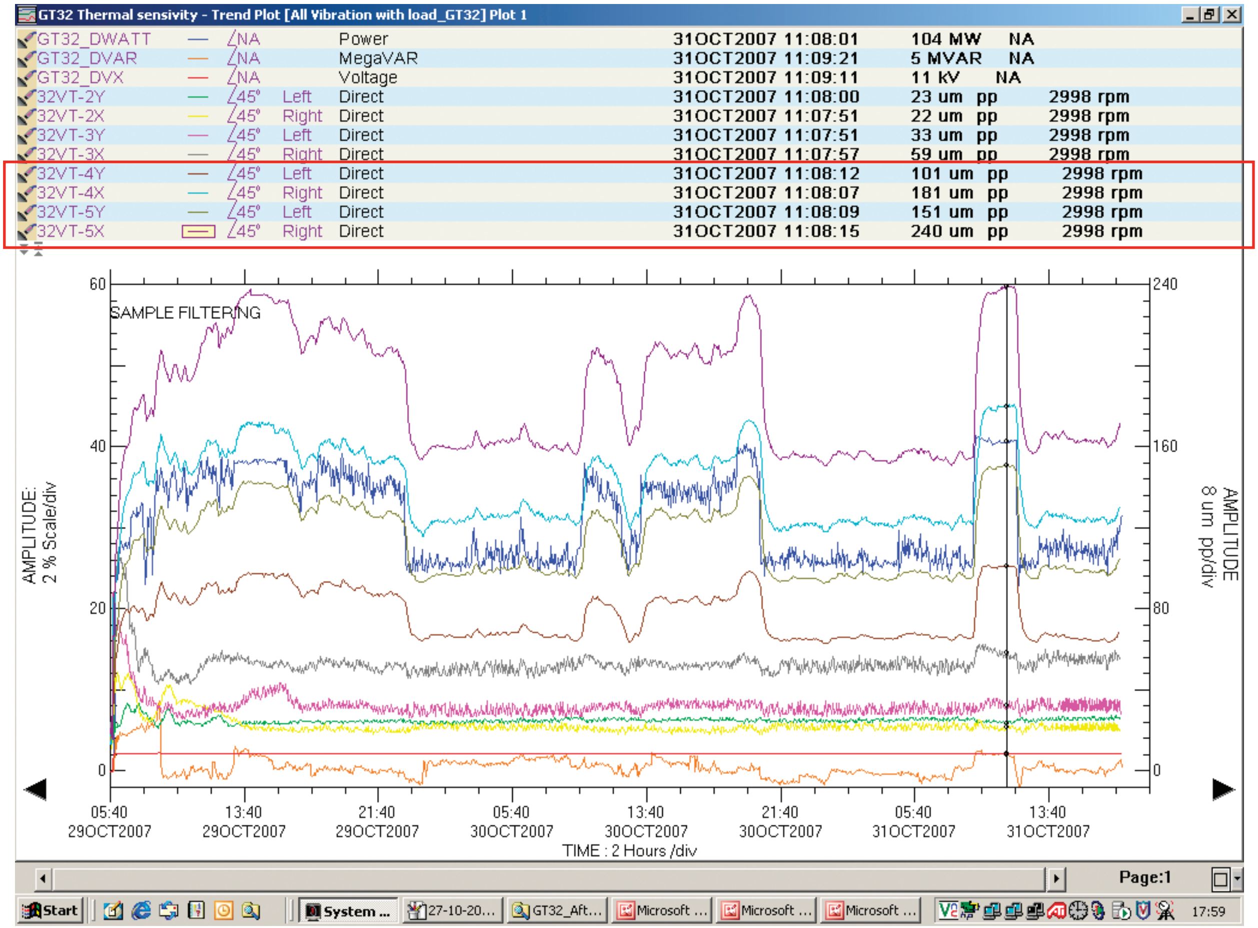
- Eddy current displacement transducers for measuring shaft vibration at all gas turbine and generator bearings (five measurement planes in total) plus a Keyphasor* transducer for measuring vibration phase angle and machine speed.
- A continuous online machine condition monitoring system including a Bently Nevada* 3500 Series vibration monitoring system with Transient Data Interface (TDI) module for communicating with a Bently Nevada System 1* condition monitoring and diagnostic system.
- An EfficiencyMap* online performance monitoring system for equipment and plant thermodynamic monitoring and optimization. (The System 1 database is used as the plant historian data server for the performance monitoring calculation engine.)
- The existing GE Mark** IV machine control system was upgraded to a Mark VI system. All process data was exported to the System 1 server for correlating with vibration and performance calculation data using GE Standard Messages (GSM) protocol. For the steam turbine-generator set, process data were imported from the plant's distributed control system (DCS) using Modbus® communication protocol.

The continuous online condition monitoring system provided value to EGAT from the very first startup during ongoing project commissioning when the on-site machinery diagnostic engineer noticed abnormal vibration characteristics at the generator measurement planes. Vibration amplitudes were very sensitive to generator MW and MVAR loads, with levels corresponding closely to generator loading.

Fluctuating vibration amplitudes were around 200 to 300 microns depending on generator load, with a predominantly synchronous (1X) vibration frequency component. Plant personnel were advised to closely monitor and investigate further for the root cause. The trend plot of generator vibration amplitudes versus load is shown in Figure 1.

Initially, a thermal sensitivity problem was suspected since the rotor's behavior appeared to be consistent with a thermal bow. As load increased, the synchronous vibration vectors at both ends of the generator remained in-phase with each other, with increasing amplitude.

This symptom tends to occur when there is a temperature gradient across the generator rotor. The higher temperature side of the rotor expands more than the lower temperature side and causes the rotor to bow when field current increases. There are many different factors that can cause thermal sensitivity of a



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GT32_DVAR     ∠NA           MegaVAR   31OCT2007 11:09:21   5 MVAR    NA
GT32_DVX      ∠NA           Voltage   31OCT2007 11:09:11   11 kV    NA
32VT-2Y       ∠45°  Left    Direct    31OCT2007 11:08:00   23 um  pp      2998 rpm
32VT-2X       ∠45°  Right   Direct    31OCT2007 11:07:51   22 um  pp      2998 rpm
32VT-3Y       ∠45°  Left    Direct    31OCT2007 11:07:51   33 um  pp      2998 rpm
32VT-3X       ∠45°  Right   Direct    31OCT2007 11:07:57   59 um  pp      2998 rpm
32VT-4Y       ∠45°  Left    Direct    31OCT2007 11:08:12   101 um  pp      2998 rpm
32VT-4X       ∠45°  Right   Direct    31OCT2007 11:08:07   181 um  pp      2998 rpm
32VT-5Y       ∠45°  Left    Direct    31OCT2007 11:08:09   151 um  pp      2998 rpm
32VT-5X       ∠45°  Right   Direct    31OCT2007 11:08:15   240 um  pp      2998 rpm
```

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240
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80
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29OCT2007
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30OCT2007
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31OCT2007
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Start
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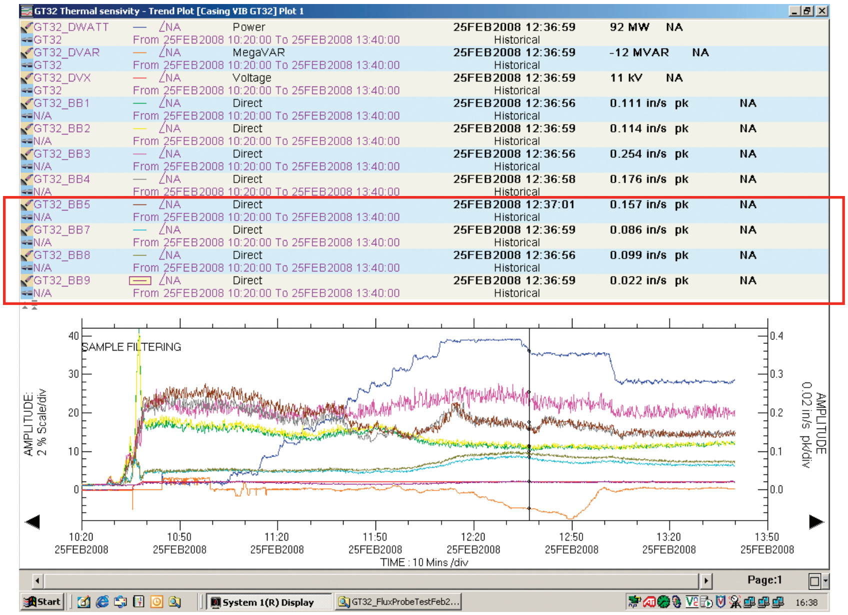
27-10-20...
GT32_Aft...
Microsoft ...
Microsoft ...
Microsoft ...
17:59

FIGURE 1: Symptoms of a thermally-sensitive rotor were observed during plant startup. The bold blue curve represents generator load, which varied between about 25% and 40% of full load. Most of the remaining curves represent vibration amplitudes, measured as relative displacement of the rotor within its bearings. Observe that these vibration amplitude curves have shapes that are closely match the load profile. The numerical values for the measured parameters correspond to data collected at the time where the trend plot cursor is located near the right end of the plot (31OCT2007, at 11:08).

generator rotor, so further testing and data analysis was required in order to pinpoint and confirm the cause.

Although generator shaft vibration (measured by displacement transducers relative to the bearings) showed a significant high-amplitude response with increased field current, there was not a significant change in seismic vibration amplitude measured on the generator bearing housings. In fact, the small observed change in overall seismic vibration amplitudes was still far below the machine manufacturer's monitoring setpoint at 0.50 in/s pk, as shown in Figure 2.

Plant operators, looking just at the machine control system – where only seismic vibration data is available – would not have noticed the load-sensitive vibration behavior. This points out an important benefit of direct shaft vibration measurement over indirect seismic measurement when the goal is to detect the movement of the shaft within the clearances of its bearings.



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GT32_DVAR    —  ∠NA    MegaVAR        25FEB2008 12:36:59   -12 MVAR    NA
GT32         From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_DVX     —  ∠NA    Voltage        25FEB2008 12:36:59   11 kV    NA
GT32         From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_BB1     —  ∠NA    Direct         25FEB2008 12:36:56   0.111 in/s  pk       NA
N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_BB2     —  ∠NA    Direct         25FEB2008 12:36:59   0.114 in/s  pk       NA
N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_BB3     —  ∠NA    Direct         25FEB2008 12:36:56   0.254 in/s  pk       NA
N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_BB4     —  ∠NA    Direct         25FEB2008 12:36:58   0.176 in/s  pk       NA
N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_BB5     —  ∠NA    Direct         25FEB2008 12:37:01   0.157 in/s  pk       NA
N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_BB7     —  ∠NA    Direct         25FEB2008 12:36:59   0.086 in/s  pk       NA
N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
GT32_BB8     —  ∠NA    Direct         25FEB2008 12:36:56   0.099 in/s  pk       NA
N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
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N/A          From 25FEB2008 10:20:00 To 25FEB2008 13:40:00     Historical
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TIME : 10 Mins /div
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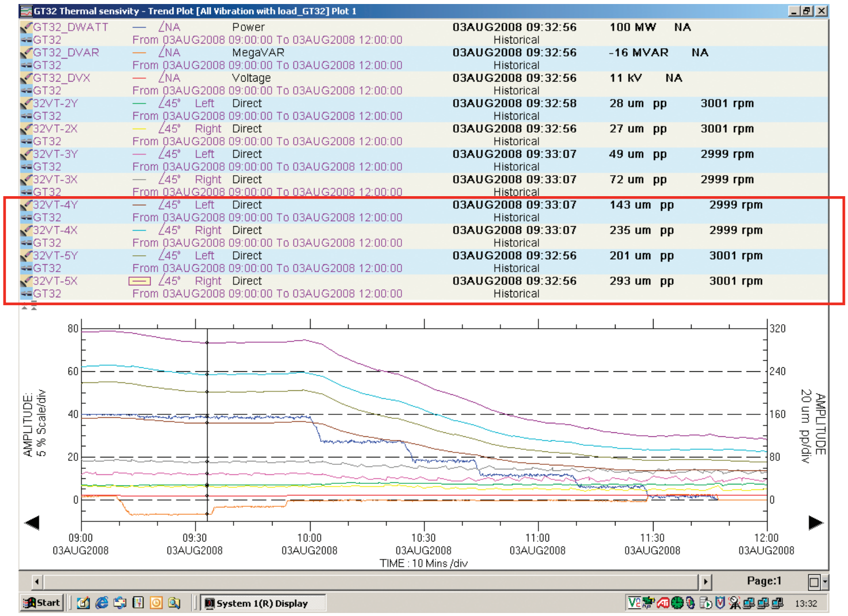
Start
System 1(R) Display
GT32_FluxProbeTestFeb2...
16:38

FIGURE 2: Seismic vibration amplitude trends from generator bearing caps, before partial rewinding repair. The bold blue "stair-step" curve represents generator load. Most of the remaining curves represent vibration amplitudes, measured as absolute velocity at the surface of the bearing housings where the seismic sensors are installed. Observe that these vibration amplitude curves do not appear to have any apparent correlation with the load profile.



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GT32 Thermal sensivity - Trend Plot [All Vibration with load_GT32] Plot 1
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GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
GT32_DVAR    —  ∠NA          MegaVAR  03AUG2008 09:32:56   -16 MVAR    NA
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
GT32_DVX     —  ∠NA          Voltage  03AUG2008 09:32:56   11 kV    NA
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-2Y      —  ∠45°  Left   Direct   03AUG2008 09:32:58   28 um  pp      3001 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-2X      —  ∠45°  Right  Direct   03AUG2008 09:32:56   27 um  pp      3001 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-3Y      —  ∠45°  Left   Direct   03AUG2008 09:33:07   49 um  pp      2999 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-3X      —  ∠45°  Right  Direct   03AUG2008 09:33:07   72 um  pp      2999 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-4Y      —  ∠45°  Left   Direct   03AUG2008 09:33:07   143 um  pp      2999 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-4X      —  ∠45°  Right  Direct   03AUG2008 09:33:07   235 um  pp      2999 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-5Y      —  ∠45°  Left   Direct   03AUG2008 09:32:56   201 um  pp      3001 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
32VT-5X      —  ∠45°  Right  Direct   03AUG2008 09:32:56   293 um  pp      3001 rpm
GT32         From 03AUG2008 09:00:00 To 03AUG2008 12:00:00     Historical
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AMPLITUDE: 5 % Scale/div
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0
320
240
160
80
0
AMPLITUDE 20 um pp/div
09:00 03AUG2008
09:30 03AUG2008
10:00 03AUG2008
10:30 03AUG2008
11:00 03AUG2008
11:30 03AUG2008
12:00 03AUG2008
TIME : 10 Mins /div
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Start
System 1(R) Display
13:32

FIGURE 3: Thermal sensitivity test before partial rewinding repair. Generator load was reduced in step intervals (again shown by the bold blue curve) about once every 20 minutes, and vibration amplitudes fell accordingly (as measured by the eddy current proximity transducers).

Thermal Sensitivity Testing

In order to confirm the suspected thermal bowing problem, the unit was subjected to a thermal sensitivity test (Figure 3) by deliberately varying generator load while closely monitoring vibration data with the online continuous machine condition monitoring system. It was found that vibration amplitudes measured on the generator were load-dependent, with amplitude increasing when generator power and field current increased and decreasing when load decreased.

The maximum shaft vibration amplitudes at normal operating condition exceeded the Zone C limit of ISO 7919-2 Standard boundary (260 microns displacement), falling within Zone D (Reference 3). Vibration amplitude in this zone is normally considered of sufficient severity to cause damage to the machine; therefore, continued operation in this zone is not recommended.

The excessive overall vibration amplitude showed a predominant synchronous frequency component due to rotor unbalance caused by a rotor thermal bow under increased generator load.

This behavior is caused by asymmetric thermal distribution across the generator rotor, resulting in uneven thermal expansion. Typically, a localized hot spot on the rotor forging tends to thermally expand more than the opposite side of the rotor, initiating the thermal bow. The rotor bow amplitude increases with increasing field current and MW load since more heat must be dissipated in the rotor at higher load operation. In the case of a rotor winding fault (such as a shorted turn) the localized heating, and hence rotor deflection, is closely related to the rotor current.

Root Cause Determination

First, the possibility of asymmetrical cooling of the generator rotor was investigated by reviewing on-line process data of hydrogen gas temperature measurement at varying loads over a period of time. This evaluation showed no correlation between hydrogen temperature and generator shaft vibration, so asymmetrical cooling was eliminated as a root cause possibility.

The investigation then focused on a generator rotor shorted turn scenario. Shorted turns in the generator rotor are commonly the results of insulation breakdown caused by aging and turn-to-turn movement. The unit was aging, with a long history of peaking service, with frequent load changes that closely match the expected profile for this failure mode. Other known mechanisms for thermal bowing of the rotor include coil foreshortening, end-strap elongation, inadequate end-turn blocking, metallic contamination, and moisture induced into the generator (Reference 4).

Coil foreshortening refers to an effect that occurs over a number of heat-up and cool-down cycles of the rotor. The copper bars in the rotor slots expand more than the steel forging of the rotor during heat-up. But friction between the bars and slots – especially near the ends of the rotor – limits the lengthwise expansion of the bars, causing the copper to become compressed (and plastically deformed, if its yield strength is exceeded). After the rotor is stopped and cooled down, the copper bars contract, pulling the end windings toward the rotor, which can cause the end windings to become misaligned and even cracked.

End-strap elongation is an effect that is caused by friction between the winding end-straps and the retaining rings. The retaining rings increase in diameter (due to centrifugal force and thermal expansion) as the generator is started and loaded. If the friction force between the top of the end straps and the insulation on the retaining rings is too high, the end straps will move with the retaining rings as they expand. If the yield strength of the copper is exceeded, permanent deformation may occur, causing them to be elongated. This cycle can be repeated over and over as the generator is started, loaded, unloaded, and stopped over its life.

Flux Probe Testing

Plant staff had a suitable period of time to investigate further during a gas turbine combustion path inspection in February, 2008. To study the suspected shorted-turns problem, a flux probe was installed and testing was carried out using

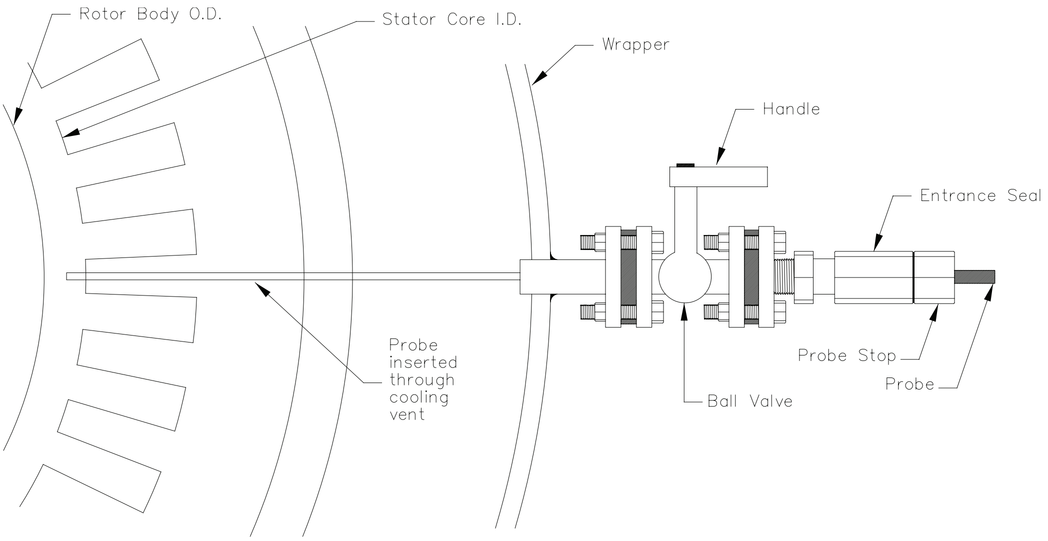
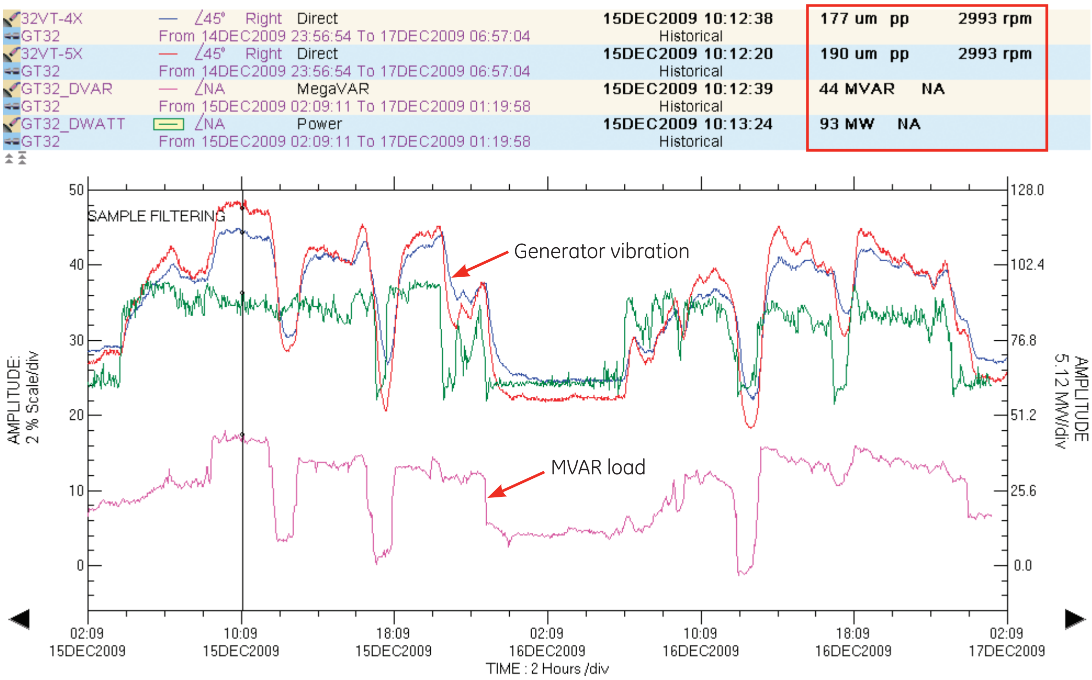

Rotor Body O.D.
Stator Core I.D.
Wrapper
Handle
Entrance Seal
Probe inserted through cooling vent
Probe Stop
Probe
Ball Valve

FIGURE 4: This simplified drawing illustrates a temporary air-gap flux probe test setup. It is often possible to install such a probe through an existing access port that allows access to the air gap through a cooling vent in the stator core. For simplicity, this drawing does not show the conductors installed in the stator slots. *Image copyright © 2012 Generatortech, Inc. All rights reserved. Used with permission.*



```
32VT-4X      —   ∠45°  Right  Direct           15DEC2009 10:12:38    177 um  pp        2993 rpm
GT32         From 14DEC2009 23:56:54 To 17DEC2009 06:57:04   Historical
32VT-5X      —   ∠45°  Right  Direct           15DEC2009 10:12:20    190 um  pp        2993 rpm
GT32         From 14DEC2009 23:56:54 To 17DEC2009 06:57:04   Historical
GT32_DVAR    —   ∠NA          MegaVAR          15DEC2009 10:12:39    44 MVAR   NA
GT32         From 15DEC2009 02:09:11 To 17DEC2009 01:19:58   Historical
GT32_DWATT   ▭   ∠NA          Power            15DEC2009 10:13:24    93 MW    NA
GT32         From 15DEC2009 02:09:11 To 17DEC2009 01:19:58   Historical
```

SAMPLE FILTERING
Generator vibration
MVAR load
AMPLITUDE: 2 % Scale/div
50
40
30
20
10
0
AMPLITUDE 5.12 MW/div
128.0
102.4
76.8
51.2
25.6
0.0
02:09 15DEC2009
10:09 15DEC2009
18:09 15DEC2009
02:09 16DEC2009
10:09 16DEC2009
18:09 16DEC2009
02:09 17DEC2009
TIME : 2 Hours /div

FIGURE 5: Generator vibration amplitude versus time after partial rewinding repair. This trend plot shows a time period of two days, with load profile showing typical variation for a non-base-loaded generation unit.

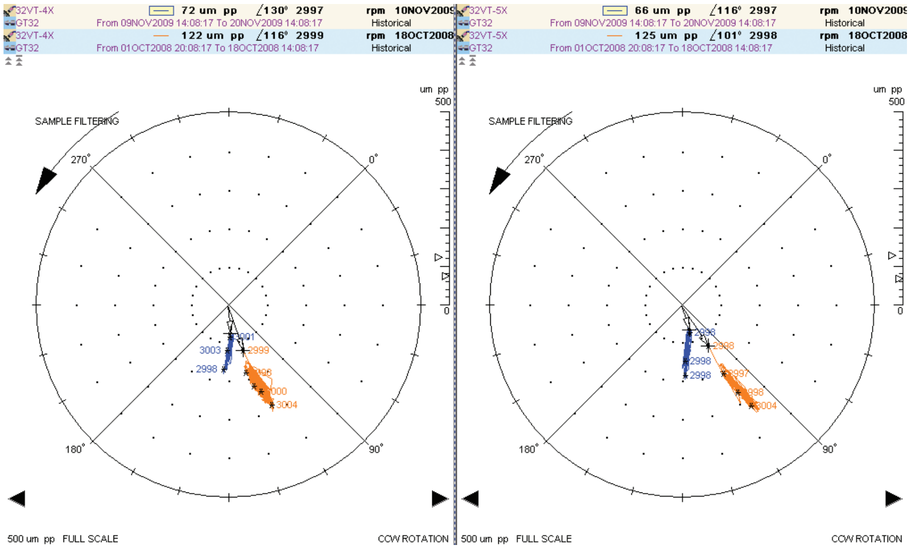


FIGURE 6: Polar plots showing generator rotor thermal vector before (orange data) and after (blue data) partial rewinding repair. Following the repair, vibration amplitudes were about half of the pre-repair values. Also, although the phase angle shifted about 30 to 35 degrees, it was still in-phase from one end of the generator to the other. The left plot shows data from bearing #4 (DE), and the right plot is from bearing #5 (NDE).

sophisticated data acquisition instrumentation and specialized data analysis techniques. The flux probe provides on-line monitoring of generator magnetic flux across the air gap when the generator is running in normal operating conditions.

This method can be used to detect shorted turns, as it is sensitive to rate-of-change of radial flux in the air gap. As each rotor slot passes the flux probe, the slot leakage flux from that slot is detected. Rotor slot leakage flux refers to flux that does not cross the air gap to reach the stator winding. Since the leakage flux does not include stator current, the leakage flux does not contribute to power generation (Reference 4). As shown in Figure 4, the flux probe can be installed through a suitable fitting on either a temporary or a permanent basis.

Indications of potential shorted-turns were found at one turn of coil #2, three turns of coil #3, and one turn of coil #7. However, the root cause could not be confirmed until a visual inspection could be carried out. At this time a physical inspection of the generator rotor assembly could not be performed due to operational constraints. The inspection had to be deferred until the next hot path inspection, which was not scheduled to occur until the final quarter of 2008.

Visual Inspection Findings

During the subsequent hot gas path inspection (October to December, 2008), a visual inspection of the generator windings was performed. The inspection confirmed the presence of failed insulation leading to the already suspected shorted-turn problem at coils #2 and #3. At these short circuited locations, electrical resistance is relatively low, resulting in low rotor temperature whereas on the opposite side of the rotor, where resistance is normal, the temperature is somewhat higher. The rotor tends to expand on the higher temperature side causing the rotor to thermally bow and subsequently causing synchronous vibration to increase (Reference 1).

Since available time during the hot gas path inspection was very limited, a full generator rotor rewind repair could not be carried out, requiring instead a partial rewind repair of the rotor. Following the partial repair, the

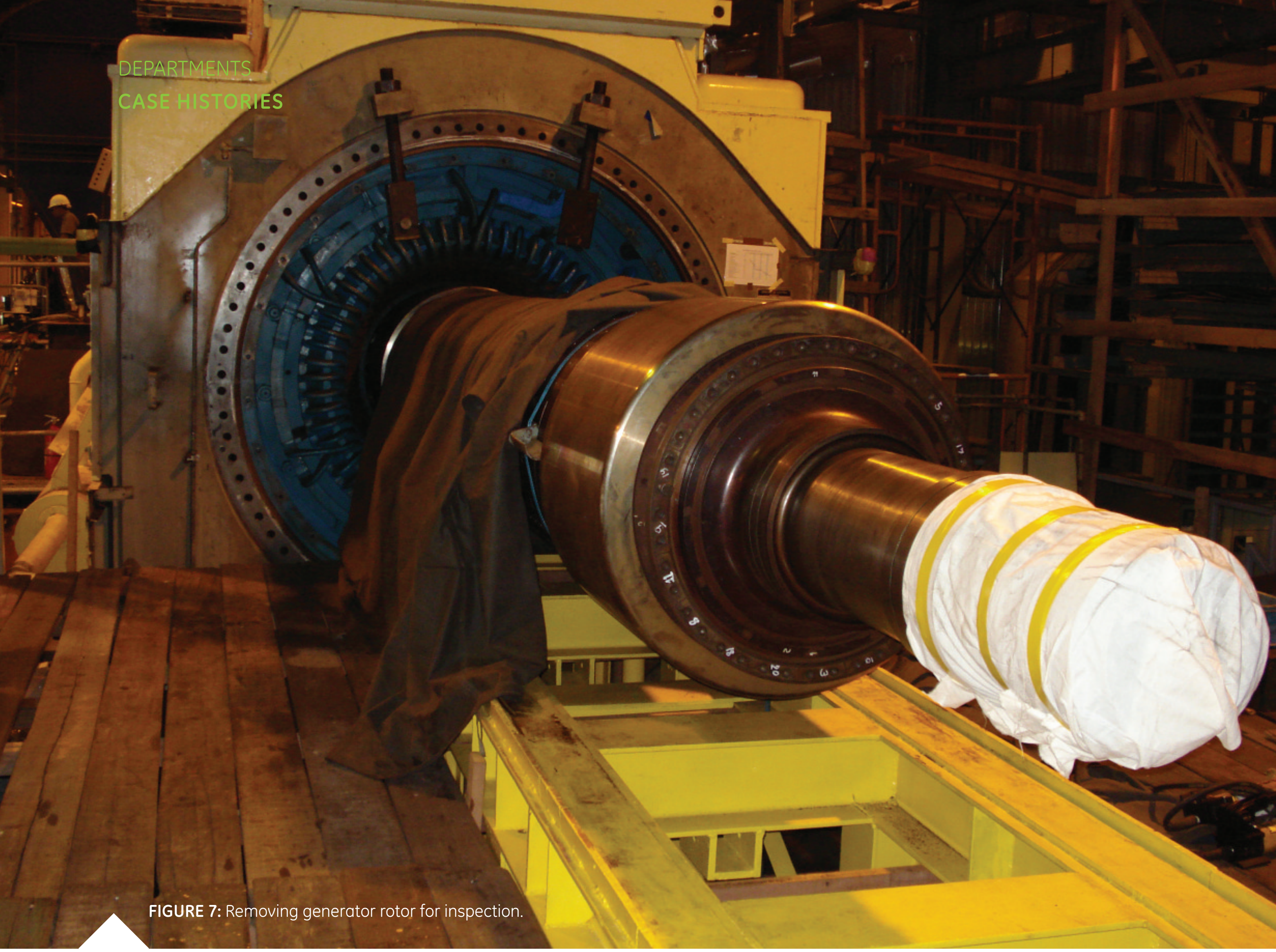


FIGURE 7: Removing generator rotor for inspection.

shaft vibration amplitude appeared to exhibit a similar thermal sensitivity as before the repair. However, the vibration amplitude had actually fallen by about 50% at comparable load operation, fluctuating at about 150-200 microns during load variation as shown in Figure 5.

Vibration amplitudes were now within Zone C of ISO 7919-2 standard, exceeding 165 microns of Zone B boundary (Reference 3), which is normally considered unsatisfactory for continuous long-term operation. Generally, the machine may be operated for a limited period of time with this condition until a suitable opportunity arises for remedial action. In this case, the unit would need to operate until the next major overhaul when a full rewind repair could finally be performed.

Following the partial rewind, the synchronous vibration vectors generally showed the same behavior as earlier, only with lower amplitudes. Phase angle was within the same quadrant as before (Figure 6). Thermal vectors at both generator ends were also found to be repeatable when comparing data over several startups. This indicated the unit was a good candidate for a "compromise balance" in order to reduce synchronous vibration at normal operating conditions. Low dynamic load resulting from improved balance could possibly extend bearing life and offer a safer operating margin.

Detailed Inspection Results

During the hot gas path inspection outage of October to December, 2008, the generator rotor was removed for a detailed visual inspection (Figure 7). Significant evidence of shorted-turns due to failed insulation was found at coil #2 and #3 as shown in Figures 8 & 9, confirming the flux probe test results. These turn-to-turn insulation failures were most likely caused by degradation from aging of the insulation material.

Other inspection results showed accumulated metallic rust at the end windings, which could have come from the generator heater and/or hydrogen gas cooler, as well as excessive moisture due to air dryer and hydrogen purifier malfunction. Evidence was also found of some movement of the end windings and insulation blocks and also of blocked cooling holes in the rotor – all of which can contribute generator rotor thermal sensitivity problems. However, the major problem in this case was considered to be the shorted-turns, hence the partial rewind repair. A full rewind repair of the generator rotor was planned for the next major overhaul.

Recommendations

It is estimated that more than 50% of installed generators worldwide are operating with some shorted-turns, but the majority of them have no problem except in the case where excessive vibration requires reduced load capacity and may lead to a forced outage for remedial work (Reference 5).

The effects of shorted-turns are rotor thermal unbalance and associated excessive vibration, higher field current required for the same load, higher operating temperature, and a decrease in power generating efficiency (Reference 4). Due to operational constraints, the unit in this case study was not able to perform a full generator rotor rewind repair during the scheduled hot gas path inspection. Therefore, it had to return to conditional operation with close supervisory monitoring.

The Bently Nevada System 1 machinery monitoring platform is ideally suited to this application as it monitors real time dynamic (waveform) vibration data as well as relevant process data. Also, when the unit undergoes regular thermal sensitivity testing, the System 1 software can be used to continuously monitor and document generator vibration response throughout the test. Available data can also be used for diagnosing any potential problem as it occurs, as well as for predictive maintenance planning.

The following recommendations were made to closely monitor the affected unit and plan further remedial actions for the next major overhaul.

- Vibration amplitudes after the partial rewind repair were reduced to 220 microns from previous 300 microns at base load of 100 MW and 30 MVAR. This behavior is repeatable and therefore can be addressed through trim balancing. A compromise trim balance was therefore recommended to further reduce the synchronous vibration amplitude and allow unrestricted operation.
- In order to promptly identify any further deterioration of rotor integrity, vibration data should be closely monitored and correlated with relevant process variables such as MW, MVAR, field current, field voltage, generator voltage, cooling gas temperature, winding temperature.
- Until the full rewinding repair is able to be performed, it was recommended that a flux probe test be conducted every three months at different load MW and MVAR to identify any further propagation of the known shorted-turns problem.
- The rust-affected gas heater and cooler should be checked for correct function and scheduled for replacement, if necessary, during the major overhaul.
- Ingress of moisture into the generator should be investigated and remedial measures implemented to eliminate the problem. The hydrogen drying system and hydrogen gas purifier should be checked for correct function. The hydrogen gas dew point should be maintained at or below 7.5 deg C when shut down (Reference 5).

Conclusions

The identified thermal sensitivity of the generator rotor does not prevent it from operation unless excessive vibration amplitudes at normal operating condition or significant capacity constraints, or efficiency losses, necessitate a forced outage. Proper diagnosis of this type of problem requires many disciplines and integration of diagnostic tools.

Online dynamic vibration data is required to correctly monitor and

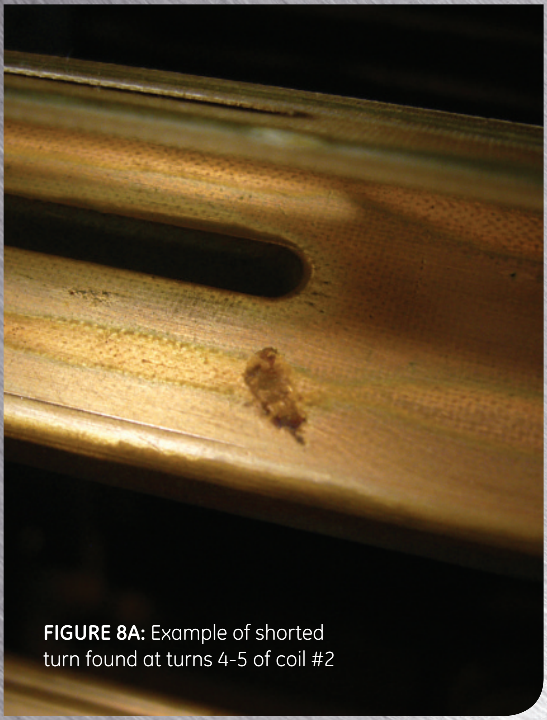


FIGURE 8A: Example of shorted turn found at turns 4-5 of coil #2

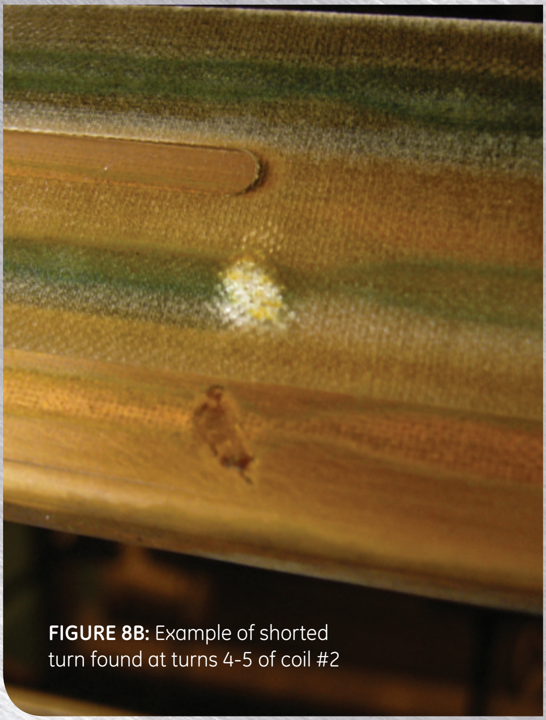


FIGURE 8B: Example of shorted turn found at turns 4-5 of coil #2

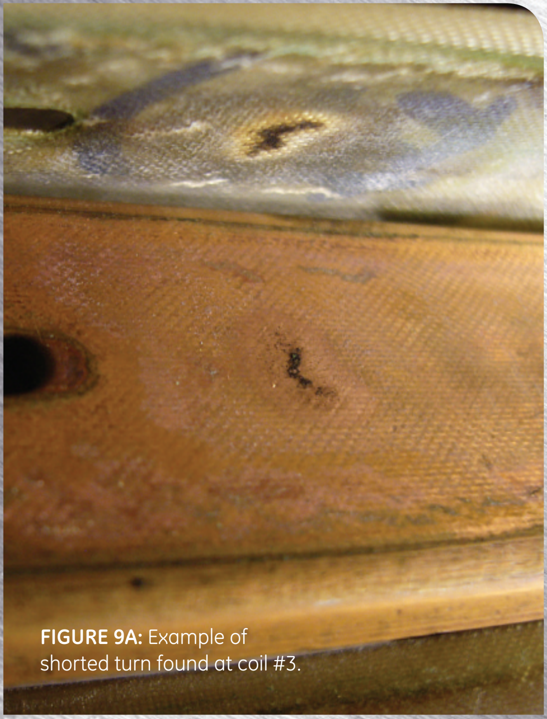


FIGURE 9A: Example of shorted turn found at coil #3.

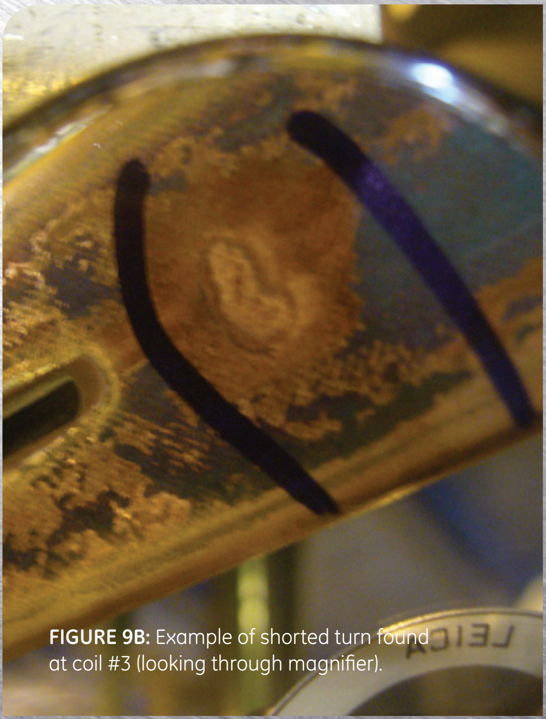


FIGURE 9B: Example of shorted turn found at coil #3 (looking through magnifier).

diagnose changes in rotor integrity over time. This includes monitoring the change in rotor deflection shape with thermal bow, dynamic vibration characteristics, and correlation with relevant process variables such as hot and cold cooling gas temperature, field current, and generation load (MW and MVAR).

Flux probe installation and associated test results can be used effectively to validate a suspected shorted-turns problem and distinguish it from other possible factors which might cause similar vibration behavior, for example, blocked ventilation, asymmetrical cooling, insulation variation, wedge fit, distance block fitting, retaining ring assembly movement, tight slots, and heat sensitive rotor forging (Reference 1).

Finally, the on-line condition monitoring system can be beneficial for real time machine condition assessment and prompt identification of any developing problems. ■

Acknowledgements

This work would not have been completed without many significant supporters. The authors would like to sincerely thank the management of EGAT Bangpakong power plant for their support and permission to publish this practical case study, and the Bently Nevada Asset Condition Monitoring team for their excellent work on this project.

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ON A FISHING TRIP IN YELLOWSTONE NATIONAL PARK

LOOKING FORWARD TO 45 MORE YEARS!

Phil Hanifan | Bently Nevada Chief Engineer (retired)

Phil Hanifan has recently retired after 45 years with the Bently Nevada team. Before he headed off to his new adventures, he left us with some nuggets of wisdom — Editor

As I consider my upcoming retirement, I look forward to many enjoyable years in that new endeavor – so I am reluctant to look back, preferring the anticipation of the unknown rather than reliving the past.

However, sometimes it is worth making an exception to that rule, and I would like to review some of the pleasantries of my decades of work experience with the Bently Nevada team. That work experience was (and still is) based upon several simple but profound principles that were first presented at a seminar at the Harvard Business School; Safety, Quality, Timeliness, Cost.

1. **Safety** – Keep the safety of yourself, your people and your customers as your first priority.
2. **Quality** – Your customers deserve and expect your absolute best performance.
3. **Timeliness** – Stick to your word and have an excellent say-do ratio as close to one as possible.
4. **Cost** – Eliminate waste and non-productive activity.

GE's Bently Nevada* Asset Condition Monitoring business elected to follow these principals – in this order of priority.

To elaborate on these principles with respect to design, never make a design that violates safety principles, always take the time to do the design right, spend the extra time, hours or dollars to meet your commitments (strive for a say-do ratio of 100%) and constantly eliminate wasted activities.

Bently Nevada (BN) exemplified those priorities under our original owner, Don Bently, and we have continued to do so under the management of the General Electric Company. Although these are simple principles to state, it is not quite as easy to learn how to demonstrate them in the business environment. Some past examples may be useful for illustration.

Safety

Bently Nevada employees were constantly reminded by our owner that we provide products that meet our customers' needs – not necessarily their wants. The method utilized to assure a proper product offering was to design based upon engineering principals.

One principal that we learned early in our BN history was that proper protection of light high speed machine rotors required directly measuring the shaft vibration with eddy current sensors rather than attempting to measure the motion of a massive casing and assuming that such a measurement properly represented the shaft motion.

However, a large and very important order opportunity was presented to us with a specification that precluded the use of eddy current transducers. While we were in the process of developing a proposal without eddy current probes, Mr. Bently walked into the conference room and asked what was going on. When we told him that we were contemplating a quotation without eddy current probes, we were suddenly presented with a new question: "Where are you guys working next week?"

This question was sufficient incentive for us to develop a quotation recommending eddy current probes – with casing measurements as an additional measurement, rather than the primary protection parameter. We lost that order to a competitor and Mr. Bently congratulated the team for doing the right thing. In less than a year, the customer suffered a high vibration event on the monitored machine, which resulted in complete destruction of the rotor. We received a subsequent quotation request from them to retrofit all similar installed machines with eddy current systems for protection.

Lesson Learned: It is always best to base our response to an order on sound engineering principles, even if it results in the order being lost at the time.

Quality

A process change implemented in the BN circuit board assembly process in the early 1970s inadvertently left contaminants on the printed wiring assemblies. When installed in a high humidity location, the contaminant activated – causing the integrated circuits to fall off the boards. We began getting calls from customers that these components, specifically the LM301 (general purpose) Op Amps, were rattling around in the bottom of the enclosure!

"…SEVERAL SIMPLE BUT PROFOUND PRINCIPLES…WERE FIRST PRESENTED AT A SEMINAR AT THE HARVARD BUSINESS SCHOOL; SAFETY, QUALITY, TIMELINESS, COST. GE'S BENTLY NEVADA ASSET CONDITION MONITORING BUSINESS ELECTED TO FOLLOW THESE PRINCIPALS – IN THIS ORDER OF PRIORITY."

We performed an immediate investigation, and when we discovered and eliminated the root cause, we searched our serial number records to locate every customer who purchased systems with this defect. All customers were contacted and visited by our Field Service technicians and all circuit boards were replaced free of charge. The expense was considerable but this was the performance that was expected by our owner.

Years later at a customer seminar I was talking with several customers. One of them said, "We hardly knew who Bently Nevada was when a Bently Field Technician knocked on the plant door and said 'You have a machine with a Bently Nevada vibration monitor installed on it and we are here to fix a potential problem with the monitor.' It was rarely heard of at that time that a supplier would fix known problems, much less come visit your site and fix a potential problem we didn't even know about yet. That sold us on Bently Nevada!"

Lesson Learned: What had initially looked like a business-ending mistake yielded benefits as an opportunity to prove that we stood behind the quality of our products.

Timeliness

Delivery time or completion time is easy to measure so it is often the first and worse yet, the only parameter measured. This automatically places Timeliness ahead of Safety and Quality unless those parameters are measured with equal precision, which nearly never happens. Therefore, one should never establish a timeliness measure unless equal or, better yet, greater emphasis is placed on safety and quality.

An example of placing the "em-PHA-sis" on the wrong "syl-AH-ble" was a strict measure of work center timeliness placed on the BN manufacturing line when I was Manufacturing Manager. There had been chronic problems with late work order completion which cascaded into late assembly and test work and ultimately late customer delivery. My Production Control Manager and I decided that we would have a timeliness meeting every Monday morning with the General Manager. Every work center supervisor who had late work orders would have to appear and explain the causes for the late deliveries.

After a while, the number of late work orders decreased dramatically.

However, at nearly the same time, customer complaints of "out-of-the-box" quality problems began to arise. We were perplexed by the quality problems since every circuit board was not only tested but run through a 72 hour elevated temperature burn cycle before being installed into the customer's final product. The customer's final product was then also fully tested before shipment. So why were we seeing these quality issues?

One evening at about 7:00 p.m., the Production Control Manager and I decided to see how the swing shift was doing. In the process of reviewing the board test and burn-in work center we saw three separate boxes of circuit cards. We asked the operator, "Why the three boxes?" The answer was, "Box 1 has boards that were fully tested and burned-in. Box 2 has boards that were tested and are ready for burn-in and Box 3 has boards that are not yet tested."

We noted that Box 3 contained the most boards and asked, "Will you get all of this done on this shift?" The operator said, "Yes, all boards will be moved to the next work center at the end of this shift whether they were fully tested and burned-in or not because we have a strict measurement of timeliness and no one wants to be called into the GMs office to explain a late work order."

Lesson Learned: Obviously, we found at least part of the source of our quality problems. When more emphasis is placed on timeliness than on quality, quality problems will reach your customer.

Cost

Lack of understanding of cost elements can lead to unsound decisions. A ten cent component purchased from a "bargain basement" supplier can incur thousands of dollars in cost to rectify the damage caused by that misplaced priority. Early in Bently Nevada history we frequently questioned some of the suppliers and the fact that they may not have been the lowest-priced option. However, in discussions with our owner, he made it perfectly clear that we were to use the best possible quality product and not the cheapest thing we could get our hands on.

One day we were presenting a significant contract to purchase integrated circuits from a Japanese supplier. At that time, the prevailing perception was that Japanese quality was very low (this was prior to the Japanese industries adopting the quality principles taught by W. Edward Deming and other world renowned quality leaders). Mr. Bently immediately called us in his office and questioned us, saying, "Why are you buying this Japanese stuff? I told you I want the best quality parts!"

After we recovered from this assault, we carefully showed him the industry data and engineering studies demonstrating that the Japanese quality was actually exceeding the quality of American suppliers at that time in our history. When presented with the data, the owner recognized that he was going to have to change his outdated perspective of where the highest quality products were being manufactured. He allowed the purchasing agreement to proceed.

Other incidents where cost was considered in a different perspective involved providing Field Service to our customers. We recognized that the production output of the machinery that our equipment monitored and protected often far outweighed the cost of our installed monitoring products, and that if our products were not fully operational, we could delay the monitored machinery from going on line.

More than once, we received an urgent customer call for Field Service support to get our equipment installed, calibrated and on-line, but there was not time to process a customer purchase order for the requested service. The owner directed us to "do what is right for the customer" and added that if we did our job right, getting a purchase order after the fact was going to be easy.

Lesson Learned: This proved to be the case in many instances, establishing a high degree of trust and confidence in the work of Bently Field Service organization and resulting in many repeat orders. ■

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COLLECTING SLOW ROLL REFERENCE DATA

Michael Thevanh
Bently Nevada Field Application Engineer (FAE)
michael.thevanh@ge.com

DEAR SYSTEM 1 USER,

In this issue, we'll show you how to collect slow roll reference data. Slow roll data is collected from proximity probes observing a rotating shaft when the rotational speeds are too slow to contribute any meaningful rotor-dynamic effects in the signal. Using slow roll reference data can be helpful when diagnosing a machine issue. It lets you remove unwanted runout to get a better picture of the machine at running speed, but this technique should be used with caution. At the end of this article you will find a reference to an Orbit article describing the use of slow roll data in detail.

We hope you enjoy this issue!
Sincerely,

Your FAE Team

3500
Trendmaster DSM
System 1
Optimization and Diagnostic Platform
GE Energy

Tip Applicability

System 1 Versions

- Applies to all versions.

System 1 Features

- Applies to System 1 Plots and Tools

Recommended User Level

- Power User
- Diagnostic User
- Mid Level User

Collecting Slow Roll Reference Data

Note: To collect slow roll data, the machine should be operating at a speed where vibration due to the forces acting on the rotor is negligible. This speed may be different depending on the machine train. For machines with turning gear systems, slow speed operation on the turning gear is the most typical time that slow-roll data is collected.

1. Open the System1 Enterprise and Display (Figure 1).
2. Select the Collection Groups tab (lower right corner of the screen)
3. If Collection Groups are not visible, select "Collection Groups" in the View menu.
4. Select the Collection Group that you want to collect the slow roll data for and right click to open the shortcut menu.
5. Select the "Collect Reference Data" option from the Data Collection menu. The "Collect Reference Data" dialog will open (Figure 2).
6. Give your dataset a meaningful Dataset Name (No spaces or special characters are allowed) and Description
7. In the "Mark Sample As" section, select the Plotting Default to designate the selected sample as the default reference that plot will use when compensating or overlaying waveform. The System Default is selected if you will be using this reference dataset in Decision Support.
8. Click **OK** to save your settings.

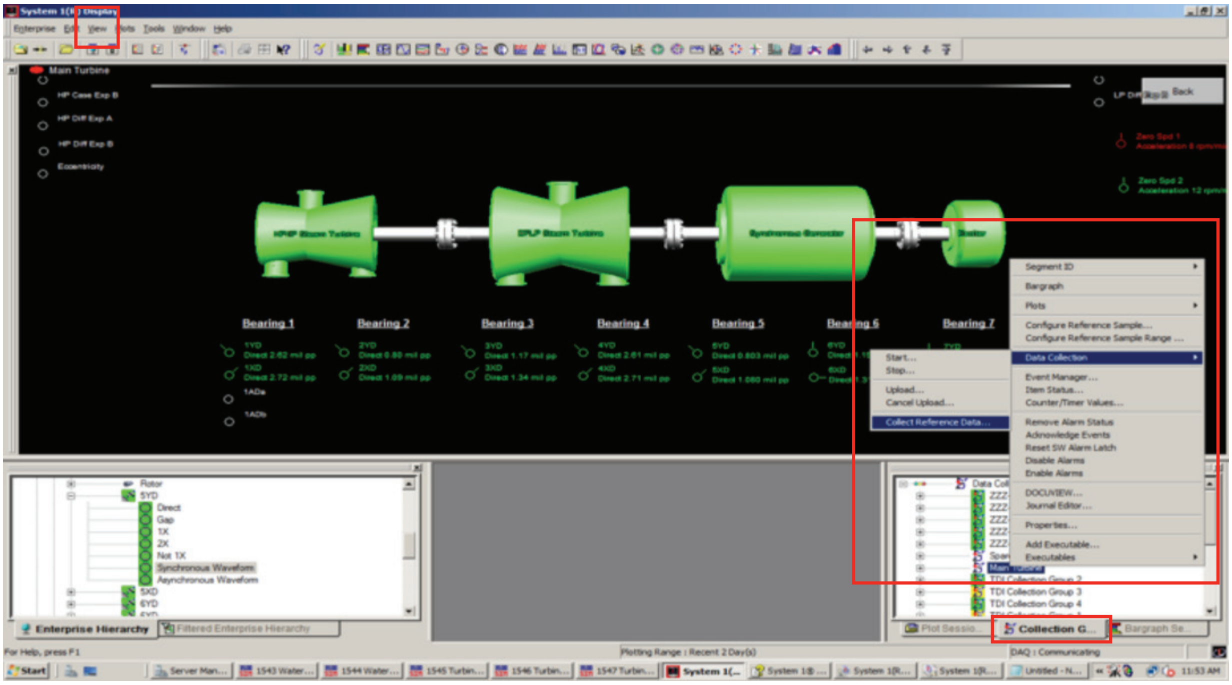
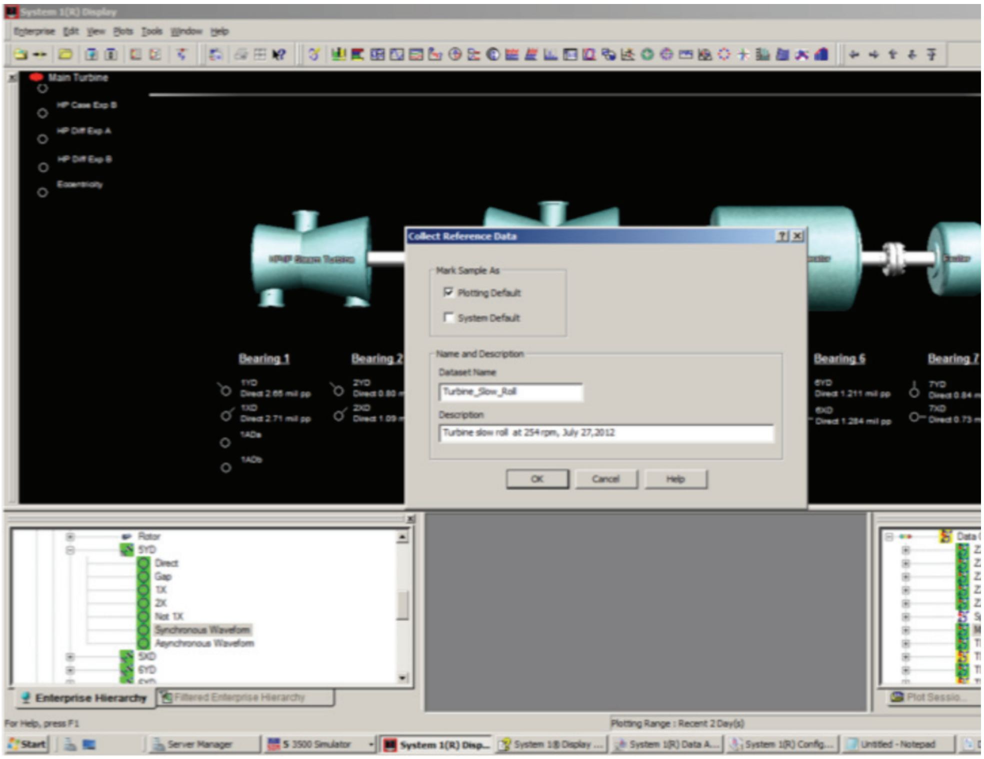

System 1(R) Display
Enterprise
Edit
View
Plots
Tools
Window
Help
Main Turbine
Bearing 1
Bearing 2
Bearing 3
Bearing 4
Bearing 5
Bearing 6
Bearing 7
Zero Spd 2
Acceleration 12 rpm/m
Start...
Stop...
Upload...
Cancel Upload...
Collect Reference Data...
Segment ID
Bargraph
Plots
Configure Reference Sample...
Configure Reference Sample Range ...
Data Collection
Event Manager...
Item Status...
Counter/Timer Values...
Remove Alarm Status
Acknowledge Events
Reset SW Alarm Latch
Disable Alarms
Enable Alarms
DOCUVIEW...
Journal Editor...
Properties...
Add Executable...
Executables
TDI Collection Group 2
TDI Collection Group 3
TDI Collection Group 4
Enterprise Hierarchy
Filtered Enterprise Hierarchy
Plot Sessio...
Collection G...
Bargraph Se...
For Help, press F1
DAQ : Communicating
Start
11:53 AM

FIGURE 1: System 1 Display screen


System 1(R) Display
Collect Reference Data
OK
Cancel
Help

Mark Sample As
☑ Plotting Default
☐ System Default

Name and Description
Dataset Name: Turbine_Slow_Roll
Description: Turbine slow roll at 254 rpm, July 27,2012

FIGURE 2: "Collect Reference Data" dialog box.

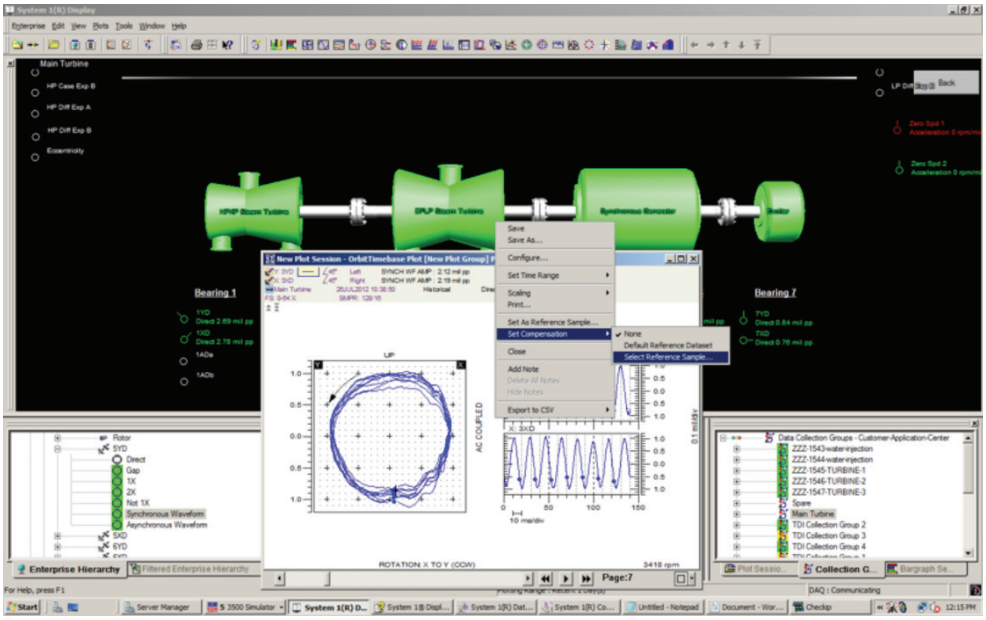

System 1(R) Display
Main Turbine
Bearing 1
Bearing 7
New Plot Session - OrbitTimebase Plot [New Plot Group]
Save
Save As...
Configure...
Set Time Range
Scaling
Print...
Set As Reference Sample...
Set Compensation
✓ None
Default Reference Dataset
Select Reference Sample...
Close
Add Note
Delete All Notes
Hide Notes
Export to CSV
ROTATION X TO Y (CCW)
Enterprise Hierarchy
Filtered Enterprise Hierarchy

FIGURE 3: Orbit–Timebase plot showing Compensation menu.

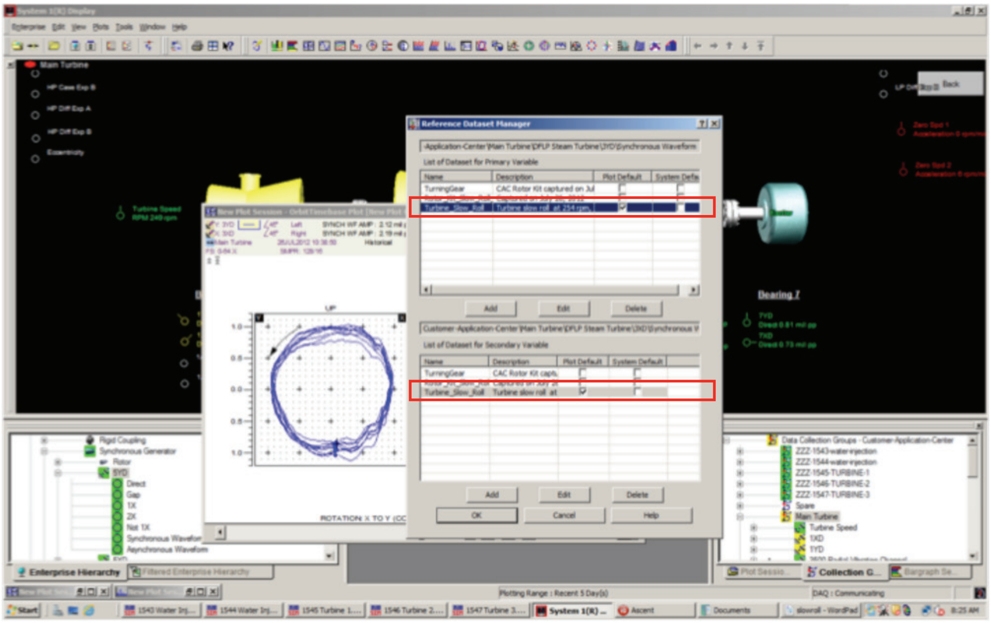


FIGURE 4: Reference Dataset Manager dialog box, showing selection of slow roll data to be used for compensation.

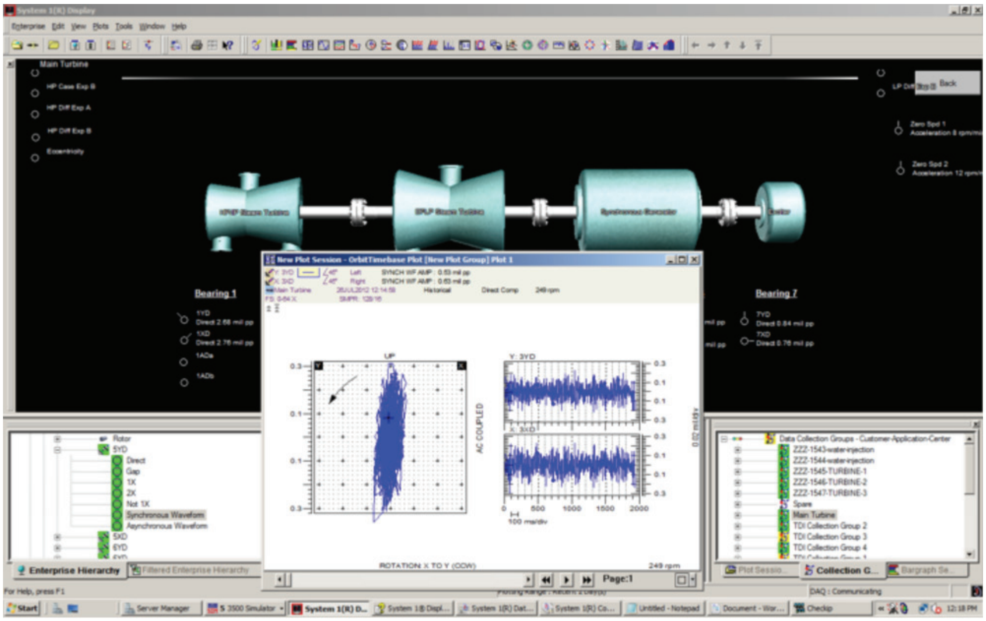


FIGURE 5: Example of compensated Orbit-Timebase plot, using the selected slow roll data. Note that this example was intentionally exaggerated to show the effects of compensation, and is not representative of a typical compensated plot.

Note: It is possible to obtain slow roll data from a historical transient (machine startup or coast-down) file if the machine cannot be shut down to capture slow roll data while it is running on the turning gear. These steps will be described in a future article

Using Slow Roll Data for Compensation

1. Open an Orbit Plot (Figure 3) for the machine of interest.
2. Right click in the plot and select Set Compensation, Select Reference Sample from the shortcut menu. This will open the Reference Dataset Manager dialog (Figure 4).
3. In the "List of Dataset for Primary Variable" section, select the appropriate reference sample, and select the Plot Default check box.
4. For paired and full spectrum plots, you can specify a Reference Dataset for each primary (Y Probe) and secondary (X Probe) variable.
5. Click OK to apply your settings to the plot. The dialog box will close and the plot will once again be visible (Figure 5). ■

Reference

Orbit Vol. 27 No. 2, 2007. Maalouf, Mel, "Slow-Speed Vibration Signal Analysis."
http://www.ge-mcs.com/download/orbit-archives/orbit_v27n207_slowroll.pdf

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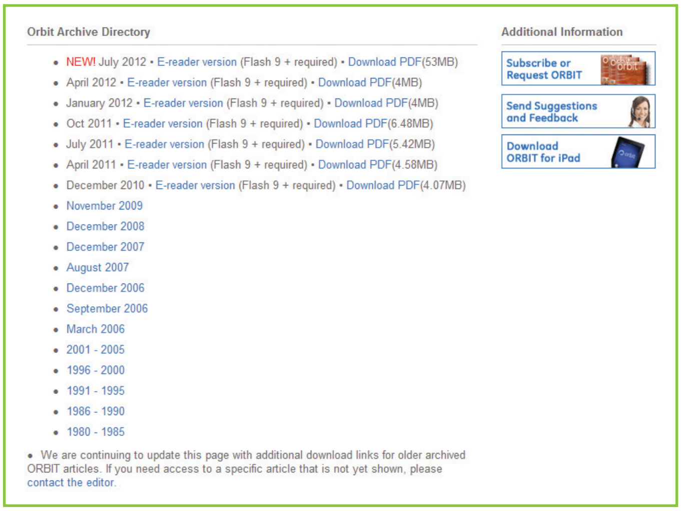

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